DESIGN AND DEVELOPMENT OF A NOVEL ELECTRO-HYDRO-MECHANICAL STEPPING MOTOR

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CERTIFICATE

This is to certify that the thesis entitled , " DESIGN AND DEVELOPMENT OF A NOVEL ELECTRO - HYDRO - MECHANICAL STEPPING MOTOR " which is being submitted by Mr. Bibhash Ghosh to the Mechanical Engineering Department of Indian Institute of Technology, Kanpur in partial fulfillment for the award of Master Of Technology is a record of bonafide work carried out by him in the department of mechanical engineering , I.I.T Kanpur under my supervision and guidance.

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CONTENTS

NOMENCLATURE

A	Cross-sectional area of piston
A_P	Rectangular port area
a	Length of rectangular port
B	A constant
b	Width of rectangular port
C	Distance between two successive rectangular ports of the valve sleeve
C_K^N	Ratio of the sum of the arms of K acting forces to the radius of cylinder locations
C_K^{NP}	A constant
d	Diameter of piston
d_V	Diameter of rotary valve
F	Resultant force transmitted from the nutating to the rotating gear
F^*	Resultant force of the axial and radial components
F_a, F_r, F_T	Axial, radial and tangential components of the resultant force
F	Viscous force
f	Stepping frequency , sec^{-1}
G_n	Force developed by a pressurized cylinder
G & G_V	Horizontal and vertical components , respectively
G_K^N	Summation of forces developed by K pressurized cylinders
h	Valve land width
i	An integer showing index of the cylinders pressurized

	in a particular step
J	Polar moment of inertia of drum
K	Number of pressurized cylinders in one step
K^{*}	Optimum number of pressurized cylinders in one step
K_{e}	Even k
K_{o}	Odd k
L	Length of the rotary valve
M_{k}^{N}	Moment caused by k pressurized cylinders on to the nutating gear in the paper plane
M_{k}^{NX}	Moment around OX axis
M_{k}^{NY}	Moment around OY axis
N	Total number of cylinders used in the EHM-SM
N_{e}	Even N
N_{o}	Odd N
n_{f}, n_{N}, n_{R}	Number of teeth of fixed,nutating and rotating gears respectively
P_{s}	Supply pressure
P_{1}	Pressure in the forward chamber
P_{0}	Pressure in the backward chamber
P_{N}, P_{R}	Circular pitches of the nutating and rotating gears
δP	Difference in pitches of the gears
Q_{k}^{N}, Q_{k}^{NH}	Average theoretical flow rate required by the motor in full step and half step regimes respectively
$Q_{k}^{N\Phi}, Q_{k}^{NH\Phi}$	Theoretical instantaneous flow rate required by the motor in full step and half step regimes respectively
$\bar{Q}_{K}^{N}$	Nondimensional average theoretical flow rate
q_{k}^{N}	Volume of flow in one step
q_{Φ}	Instantaneous flow rate when only one cylinder is

pressurized

r	Radius of pitch circle locating the cylinders
r_1	Average pitch circle radius
r_2	Mean radius of rotating gear
S_k^N, S_k^{NH}	Effective axial travel of pressurized pistons in full step and half step respectively
T_L	Frictional load torque acting on the output shaft
T_K^N, $\bar{T}_K^N$	Theoretical instantaneous torque developed by the motor in dimensional and non-dimensional forms
T_K^{NP}	Torque as a function of supplied pressure
$\bar{\bar{T}}_K^{NP}$	Non-dimensional holding torque
$T_N^{N\theta}$, $\bar{T}_N^{N\theta}$	Dimensional and non-dimensional torque as a function of pitch cone angle
$T_K^N\Big\vert_{\Phi = \alpha}$	Holding torque of the motor
$T_k^N\Big\vert_{max}$, $T_k^N\Big\vert_{min}$	Maximum and minimum torque developed by the motor
δT_k^N or δT^N	Coefficient of torque fluctuation
t	Time
V_Φ	Piston velocity
V_1	Allowable fluid velocity in the rectangular port
W	Surface area of the rotary valve
(X_i, Y_i)	Coordinates of the pressurized cylinders at the end-of-step in XOY frame
(X_i^Φ, Y_i^Φ)	Instantaneous coordinates of the pressurized cylinders
Z_Φ	Axial travel of piston as a function of Φ
δ_c	Radial clearance between valve and sleeve

α	Input step angle
β	Output step angle
γ	Tilt angle
$\bar{\gamma}$	Pressure angle of gears
θ	Pitch cone angle of the gears
ψ_1	Valve land angle
μ	Dynamic viscosity of the working fluid
ρ	Mass density of the working fluid
τ	Shear stress of the fluid
ω	Angular velocity of the rotary valve

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ABSTRACT

The objective of the present work is to design and develop a new generation of electro-hydro-mechanical stepping motor. This type of stepping motor has high resolution and can develop tremendous amount of torque. High resolution and built-in-step accuracy has been achieved with the help of fixed-nutating-rotating bevel gear combination. Hydraulic power-pack has been used to augment the developed torque. High resolution and greater amount of torque developed provides it an edge over conventional electrical stepping motor. Principle of operation and the theory behind it has been presented in a more or less lucid manner. The primary steps of design procedure has also been presented. The performance of the proposed EHM-SM is assured with conducting two primary experiments and several hours of demonstrative running. The proposed motor is capable of producing step angle equal to 0.2445 degree and torque equal to 840 kgf-cm.

CHAPTER - 1

INTRODUCTION

1.1 Definition Of Stepper Motor.

A stepping motor is a device which is capable of converting input pulses (electrical,hydraulic,mechanical) into analog output motion in the form of angular step increment. It is basically a digital to analog converter. It rotates in a series of discrete steps or angular increments in response to a series of input pulses, the rate of input pulses governing the output speed while the total number of pulses govern the final angular position.

One input pulse provides one output step increment and fixes the position of the output shaft precisely. This one-to-one correspondence of the output shaft position to input signal resembles the feedback of a conventional position servo. Perhaps the most important functional distinction is the inherent stability of a stepping motor, within certain load conditions, as compared with a stepless feedback device [1].

1.2 Application Of Stepper Motor.

Practically stepping motors are the ones that have been used in the field of digital control, especially in numerically controlled machine tools. Two very strong reasons are that they are suitable to the advancing electronic data techniques and very high stepping rates, up to 20000 pulses per second have been achieved.

However, even though the best developed fluidic controlled stepping motors will have a maximum stepping rate of less than 1000 pulses per second, there are several practical applications where electrical systems can not or should not be used because of (1) the environmental conditions are such that electrical systems

will misbehave, (2) a fluidic system better suits the available power sources, (3) the sensing devices are fluidically related and simple, thus, eliminating expensive interface devices, (4) a fluidic system is more reliable and requires less maintenance.

Stepping motors can be used in many ways and five important stepper functions are namely when using the stepper motor as an integrator, a finite summing device, a velocity servo, a position servo and a data quantizing device [1].

When used as an intregator the time-variant nature of the system is such that no significant loss is suffered regarding the stepper as a continuous function device. Many systems operate with a data influx of a very low time-variant nature, e.g. some machine processes and many chemical processes, where stepping motors are ideal intregators because of their simplicity and fine incremental accuracy.

In systems involving computer controlled actuators a bidirectional stepper may serve as a highly accurate summing device.

Used as a velocity servo, an analog function is converted to proportional pulse rate and it responds linearly to the pulse rate,within the limits of its stepping rate range.

Similar to conventional position servos with a position feedback loop and a comparator, the stepper becomes compatible with analog controls.

It can also be used in opening and closing the valve. Further, regarding the fluidic controlled stepper motor, the elimination of electrical signals in the output unit would be a safety feature where the fire hazard is great [2].

1.3 Classification Of Stepping Motors.

Stepping motor is classified in the form of a block diagram as shown in figure 1.1. Besides ,SM can also be classified based on resolution (step size), stepping speed and output torque.

Based on resolution, SM can be classified into three groups:

1.Low resolution.

2.Medium resolution.

3.High resolution.

Based on stepping speed and output torque, SM can be classified into three groups:

1.Low torque, high stepping speed. e.g. electrical stepping motor.

2.High torque, medium stepping speed. e.g. electro-hydraulic and electro-mechanical stepping motor.

3.High torque, low stepping speed. e.g. fluidic stepping motor.

Our objective is to design an electro-hydro-mechanical stepper motor which will have high torque and high resolution simultaneously.

1.4 Review Of Previous Work.

The commonest form of SM is electrical type. It works on the principle of electro-magnetic induction. This type of motor possesses low torque and high stepping speed. Pure mechanical stepping motor does not exist in practice. But feed mechanism (ratchet and pawl) in shaping machine may be considered as an example of mechanical stepping motor.

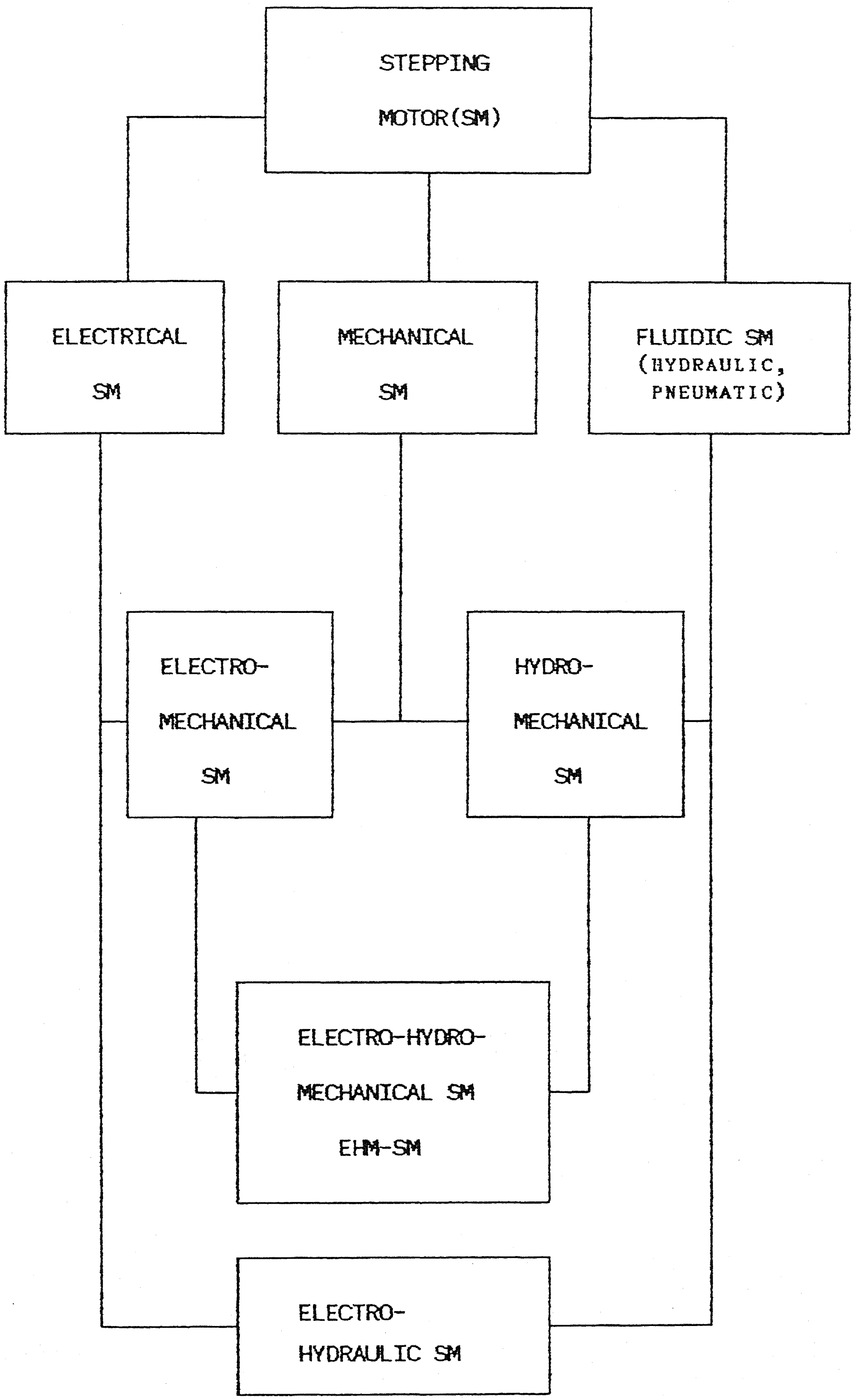


Figure.1.1 Classification Of Stepper Motor [19].

Among the electro-mechanical stepping motors, one important version is Responysm step motor [Figure 1.2]. It consists of a stator in the form of a circular spline and a flexispline. As the poles of the stator are energized in succession, the teeth difference being one, the radial force rotates and in turn steps are generated. Resolution ranges from 0.75^0 to 0.18^0, but holding or stall torque is limited [3]. One fluidic stepping motor, called Rotellmotor [Figure 1.3] works on the same principle except that the radial force is created by pneumatic actuators in the form of air bags [1]. It has the same range of resolution as that of Responsyn SM, but capable of producing greater amount of torque.

Ravishankar [4] has developed a electro-mechanical stepper motor [Figure 1.4] which is double acting type. This motor consists of a double face bevel gear mounted on the output shaft and two rows of electro-magnets force nutating bevel gears on both sides to mesh with it. Shankar's motor has resolution equal to 0.45^0,holding torque equal to 10 NM. but stepping speed is low because of the mechanical links in the form of steel cables.

A few significant developments in fluidic stepping motors are given in table - 1 together with source references. Nearly all these are developed along the line of approach of electrical stepper motors. Thus, for instance, Howland's version [5] and that of Carlnas [1] both utilize as many as eight bellows or air-bags to replace magnetic poles, and complicated fluid logic (fluidic) circuitry to drive the actuators in specific sequences.

One further development in stepping motor is hydro-mechanical type. A specific model is shown in Figure 1.5.

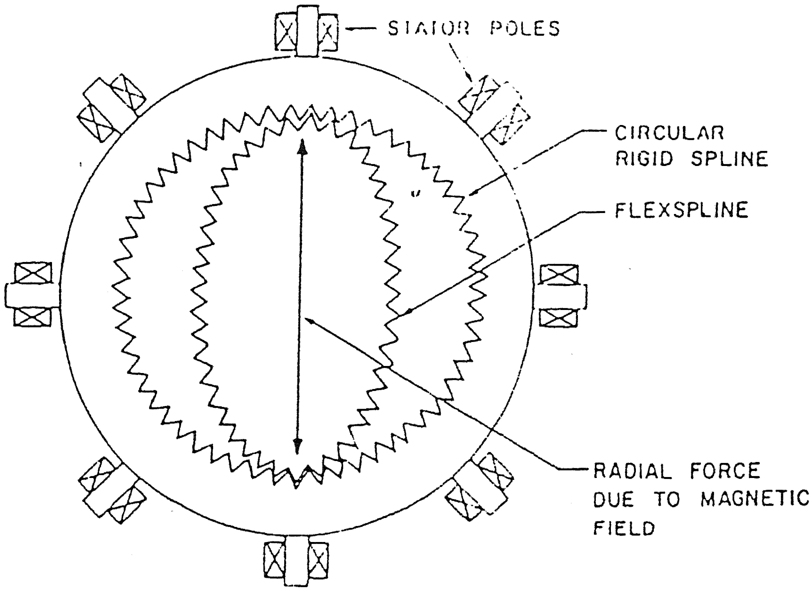


Figure 1.2 Diagram of Responsyn Motor [3]

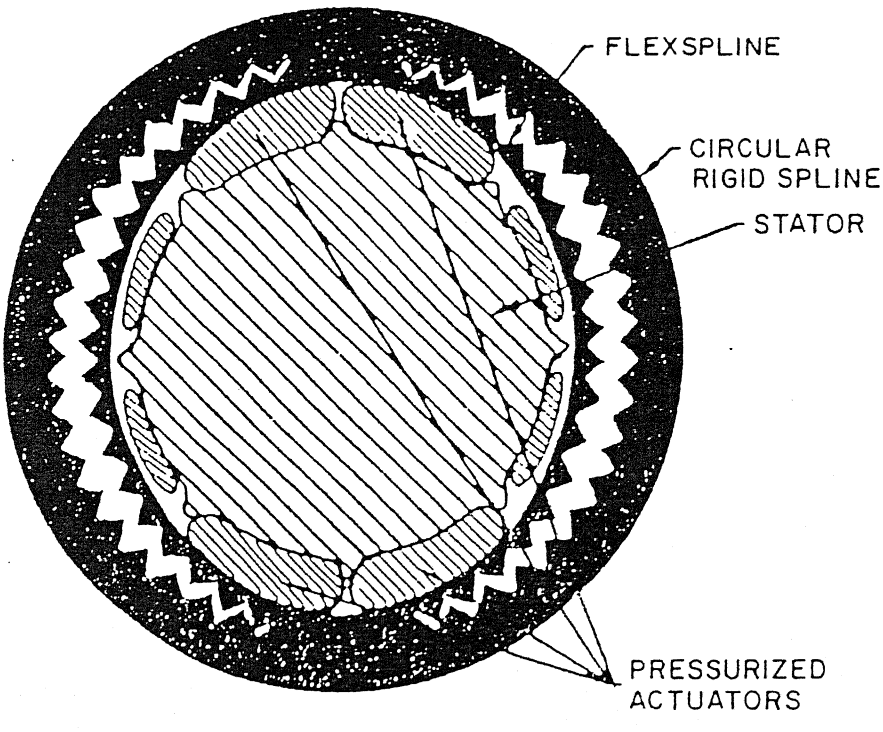


Figure 1.3 Diagram of Rotellmotor [1]

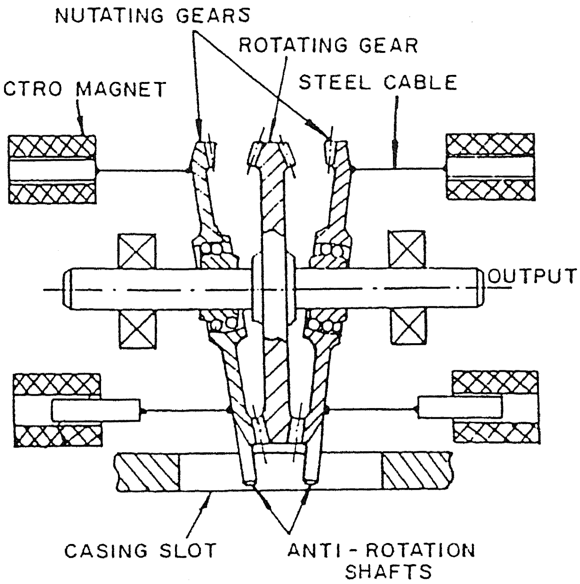


Figure 1.4 Diagram of Ravishankar's Motor [4]

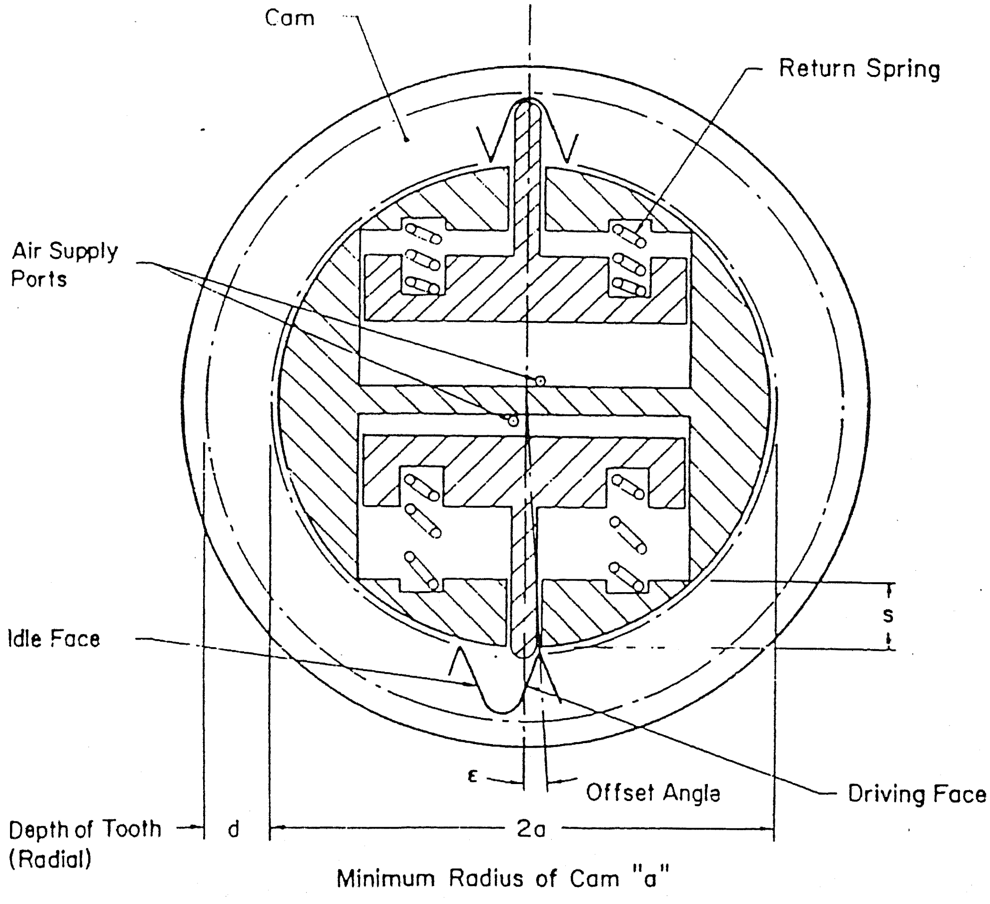


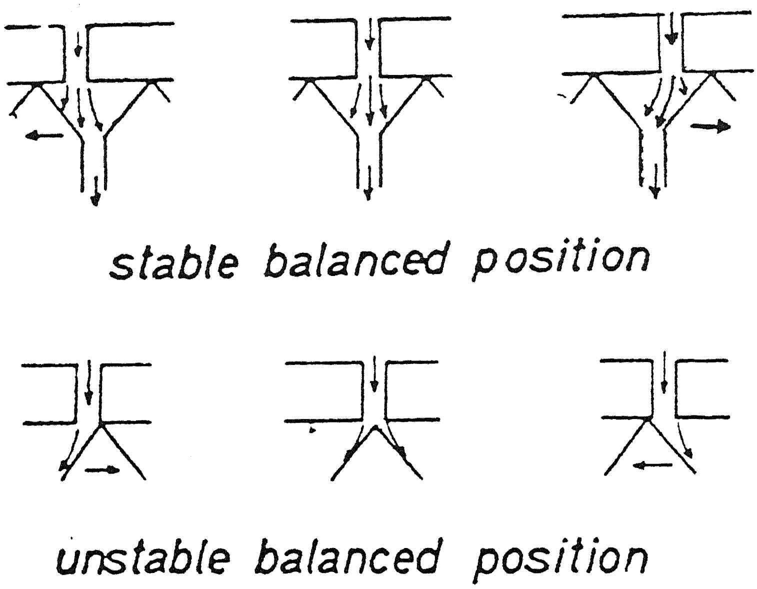
Figure 1.5 Diagram of Cheng and Fahim's Motor [21]

Reference	Operational Concept	Holding Torque N.m	Resolu-tion.	Remarks
Martin [5]	Ratchet & pawl with detent wheel	1.35	1 deg.	Design not compact due to mechanism used
Holand [6], Griffon & Cooley [7]	Gimbal-supported nutating driving gear of 181 teeth in mesh with an output gear of 180 teeth, actuated 8 bellows driven by fluidic logic	7.91	0.25 deg.	Complex design due to large no. of actuators, requires complex fluidic logic circuit
Nomoto & Shimada [8] [Figure1.6]	Utilizing impact force of an air jet	-	-	No data available
	Utilizing the reaction of an air jet	15×10^{-4}	20 deg.	Very low available torque
Blaiklock [9]	Improvements on Howlands motor	-	2.4 deg.	Complex design due to use of more bearings than the Howlands motor

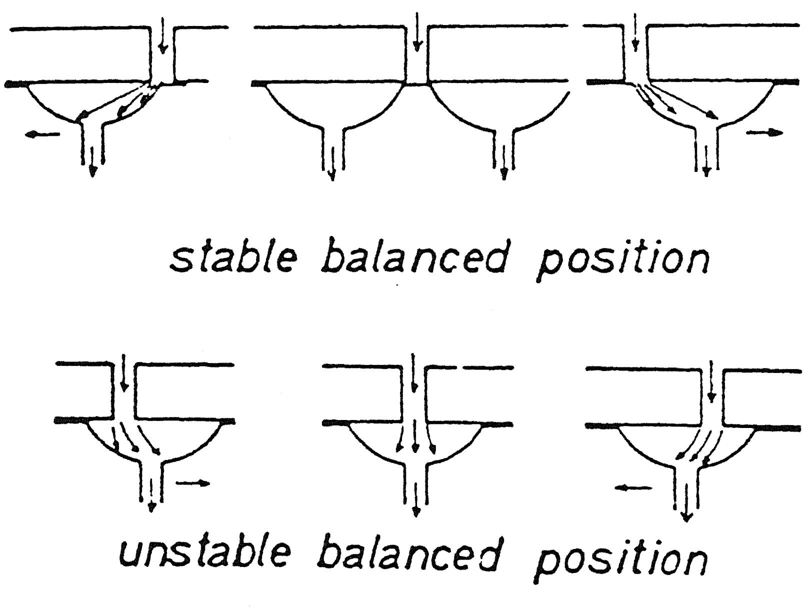
Table 1.1 Summary Of the Significant Developments In Pneumatic SM.

Hunter & Thompson [10]	Symmetrical cam with 3 sets of actuators	-	-	No data available
Warren [11]	Driven by jet acting on a pelton wheel -like rotor, and likewise stopped by another jet.	-	-	No data available
Dat,Fabre & Yalcin [12]	Wankel engine-like arrangement with special porting	1.2×10^{-4}	180 deg.	Poor performances characteristics, and poor step accuracy
	Eccentric rotor with vanes forming two chambers, operates on pressure differential across chambers	-	-	No data available
Cheng & Fahim [13 & 14] [Figure1.7]	Asymmetric internal gear-like cam with 2 collinear actuators	5.7	12 deg.	Simple design , simple control circuits, high torque and good accuracy

Table 1.1 Continued



PRINCIPLE OF IMPACT TYPE FLUIDIC STEP MOTOR



PRINCIPLE OF REACTION TYPE FLUIDIC STEP MOTOR

Figure 1.6 Diagram of Nomoto and Shimada's Motor [8]

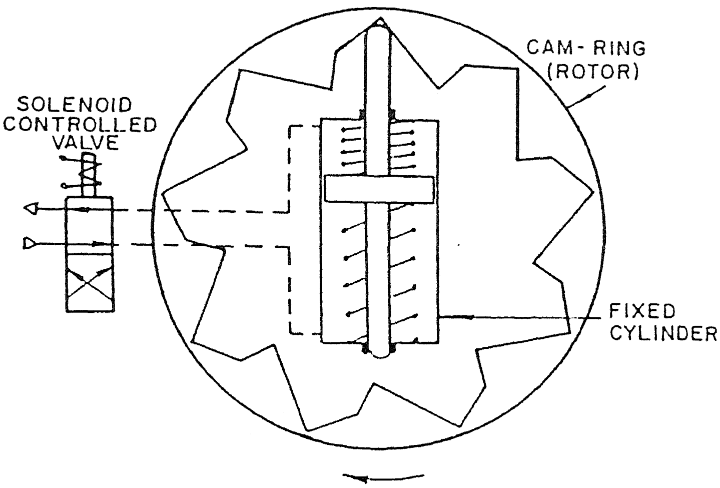
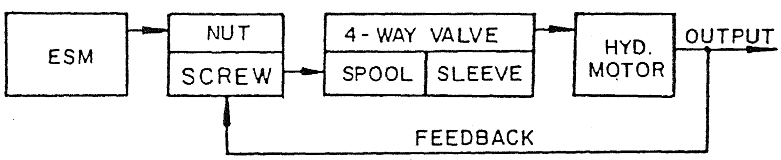


Figure 1.7 Diagram of Cheng's Motor [13 , 14]



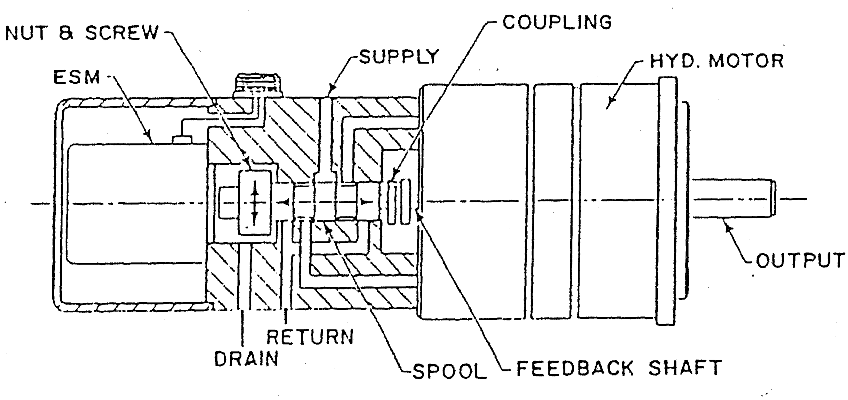


Figure 1.8 Schematic and block diagram of Motion Product made EHSM [15 , 16]

The mechanical portion of the motor, in principle, is equivalent to a double-acting cylinder with one through piston-rod. The ends of the piston-rod are designed to alternately engage and drive an asymmetric cam in a radial fashion. Thus, by means of two pneumatic 3-way valves, the actuator is made to oscillate producing one step for every half cycle of oscillation.

The model of electro-hydraulic stepping motor manufactured by Motion Product, Minneapolis, Minnesota [15, 16] is shown in Figure 1.8. The essential feature is that the hydraulic motor shaft is coupled with the pilot spool directly. The connection is rigid radially, but compliant axially. This eliminates backlash between the hydraulic motor shaft and the pilot spool. The nut and screw arrangement which is mounted on the other end of the spool acts as rotary to linear translator as well as error detecting device.

Mr. Loc [19] of I.I.T Kanpur has developed a Electro-hydro-mechanical stepper motor as shown in figure 1.9 . It includes electrical , hydraulic and mechanical devices . There is a circular array of 10 cylinders inside the cylinder block (3) in which the pistons (4) reciprocate. There are two bevel gears - (i) Nutating bevel gear (5) and (ii) Rotating bevel gear (6) . There is a rectangular slot with semi-circular ends which houses an anti-rotating shaft (13) fixed in a grooved casing of the nutating gear . So any rotary motion of the nutating gear is prevented . The fluid power is controlled by a rotary valve (14). As the pistons press the nutating gear onto the rotary bevel gear , the teeth difference being one , single step is generated .

The description of stepping motors can be continued[17,18]

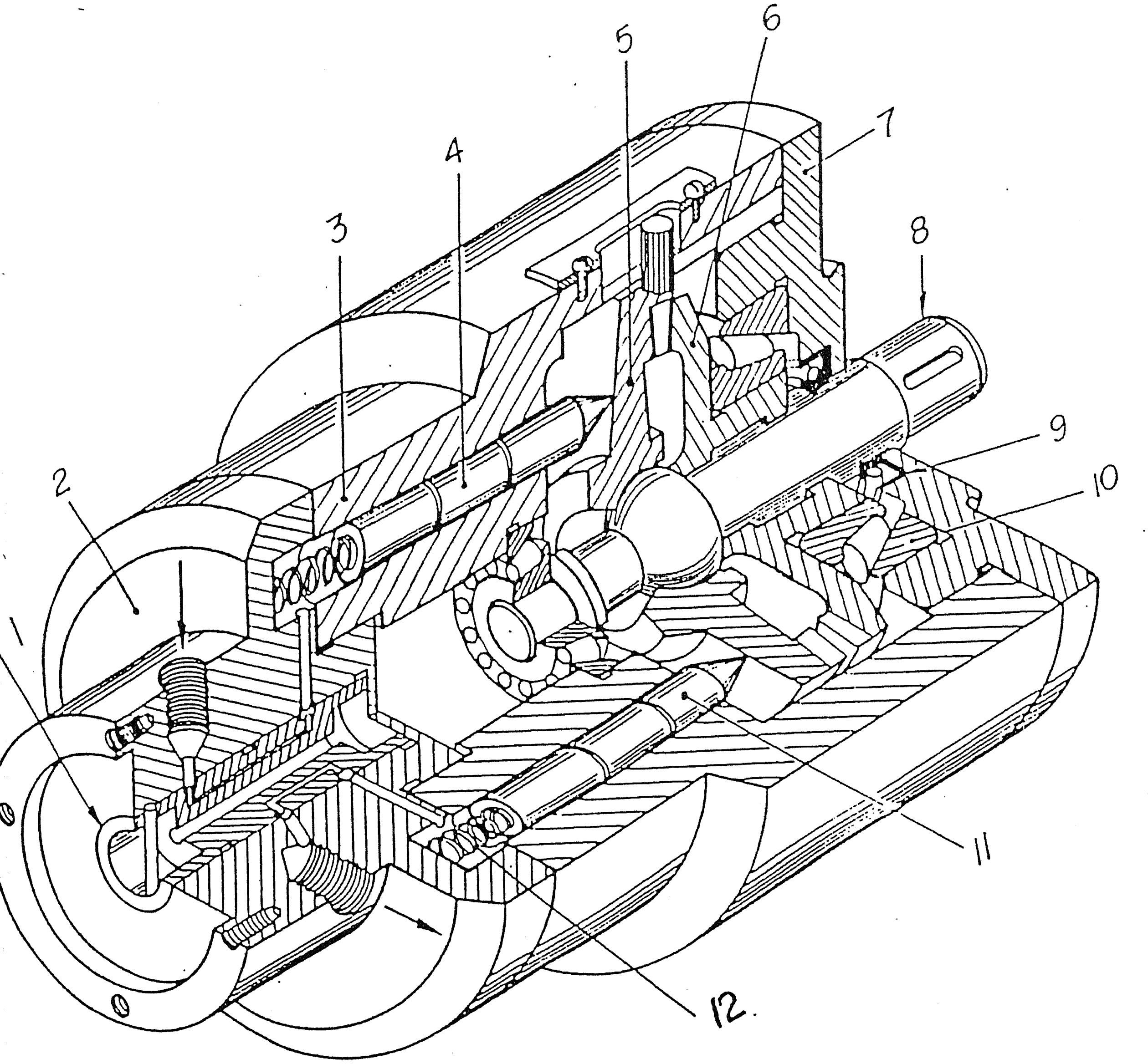


Figure 1.9 Diagram of Loc's Motor [19]

further but the basic pattern remains the same. As SM has become indispensable control device in industrial automation, efforts are on to create a new model to suit specific areas of application.

1.5 Proposed Stepping Motor.

One can conclude the following features from the stepper motors described earlier.

A. The combined SM which uses electric stepping motor as its input command pulse receiver has the following advantages:

1. High stepping speed.
2. Easy and precise control of stepping speed by controller.
3. Accepts commands from a digital computer directly.

B. The combined stepping motor which uses mechanical devices to convert electrical or hydraulic (pneumatic) energy into output step increments has the following advantages :

1. Very small resolution.
2. Built-in-step accuracy.
3. High holding torque.
4. Natural feedback between output step increments and the input command pulses.

C. The combined stepping motor using hydraulic motor as power amplification device offers the following advantages :

1. Large output torque.
2. High power to weight ratio.
3. High horse power.

Commercially available electrical stepping motor has the following drawbacks :

1. Small output torque.

2. Positional accuracy is low.

3. Low resolution (generally full step=1.8^0,half step=0.9^0)

4. No provision for changing output torque for a particular combination of controller and ESM.

5. Low power to weight ratio.

Hydro-mechanical (pneumo-mechanical) stepping motors can overcome first four drawbacks but not the last one.

Though electro-hydraulic stepper motor overcome those above mentioned drawbacks, this type of motor creates new drawbacks involving high initial and maintenance costs. Backlash in coupling and clearance between spool and sleeve are critical parameters which would adversely affect the performance. Secondly, being a close-loop control system, existing EH-SM has problems with stability of the controlled parameters, specially when working conditions such as ambient temperature and humidity change.

Therefore, we propose a new version of stepping motor known as *electro-hydro-mechanical stepping motor* (EHM-SM) which includes all the three components, namely electrical, mechanical and hydraulic devices in such a way that it can take up all the advantages of individual components and remove all the individual drawbacks. This motor has the following salient features :

a> Electrical stepping motor provides input command signals. DC Motor can also be used.

b> Fixed-Nutating-Rotating bevel gear mechanism to achieve high resolution and positional accuracy.

c> A circular array of linear hydraulic actuators.

d> Interface between ESM & hydraulic actuators through a rotary valve.

e> Open-loop principle makes the whole system simple.

Apart from that , this motor has certain specialized features that makes it a better version of Mr. Loc's motor.

1) Keeping the size of the motor same , the power output has been increased i.e better power to weight ratio .

2) Diameter and length of the valve is about half the corresponding dimensions of Mr. Loc's valve . So the torque required to rotate the valve is much less as the surface area gets reduced drastically . As the valve has been made small , care has been taken to reduce the manufacturing operations on the valve . The slots for fluid distribution from the supply source has been made on the valve sleeve instead of the valve .

3) The design is more compact . Hardly any unnecessary space can be found in the motor .

4) Anti-rotation shaft in case of Mr Loc,s motor is an asymmetric part . So fabricating of this type of part is extremely difficult. As anti-rotation shaft is screwed to the nutating gear , the joint is weak and can be considered as a poor design . Anti-rotation shaft has been replaced by fixed gear having same number of teeth as that of nutating bevel gear . This provides extremely good inherent built-in-step accuracy of the motor .

5) The hemispherical part on which the nutating bevel gear nutates has been replaced by standardized SKF self-aligning ball bearing .

6) Selection of connectors has been made in such a way that there is minimum contraction or enlargement loss of the fluid . The fluid path has been designed in such a manner that the fluid passes through the shortest possible path and without much change in direction . So there is hardly any difference between the

supply pressure and the pressure existing in the pressurized cylinders . This is in sharp contrast with Mr. Loc's motor .

7) Torque fluctuation is considerably less than that of Loc's motor in half step as well as in full step regime.

Our main objectives are as follows :

1. Development of theory of this particular EHM-SM.
2. Preliminary design of the proposed motor.
3. Fabrication of a prototype having resolution and holding torque equal to 0.489^0 and 870 Kgf-Cm.(85 Nm.) respectively.
4. Experimental study of essential characteristics such as holding torque and resolution.

CHAPTER - 2

THEORY OF PROPOSED EHM - SM

2.1 Description Of the proposed EHM-SM

The constructional diagram of proposed Electro-Hydro-Mechanical stepper motor and block diagram of the proposed motor as a system are shown in Figure 2.1 and Figure2.2 respectively.

The electrical portion of the proposed motor consists of a digital controller and an electrical stepper motor (ESM). However, DC motor with appropriate circuits can also be used for providing input rotary motion. The shaft of ESM rotates the valve (6) through a clutch (2). The output stepping speed can be varied at ease with the help of controller.

The hydraulic part of the motor consists of a rotary valve(6), a circular array of hydraulic actuators or pistons(8) and a hydraulic power package. The valve distributes fluid from the pressure source (P_s) to a group of cylinders and exhausts fluid from the other cylinders to the sump (P_0). The function of the valve is to create a predetermined pressurization pattern to the cylinders to meet the torque and step-wise motion requirements of the motor. The pistons move inside the stationary cylinder block(9) and press onto the nutating gear. The hydraulic power pack consists of a centrifugal pump, an oil sump and a distributor. The distributor is basically a 4-way valve. The input ends of the distributor are connected to the pump and the sump. The output ends are connected to forward and backward chambers of the hydraulic cylinders in the proposed motor. Forward chamber implies pressurized main port while backward chamber is the depressurized port. The fluid flow in forward and backward chambers can be interchanged by changing the positions

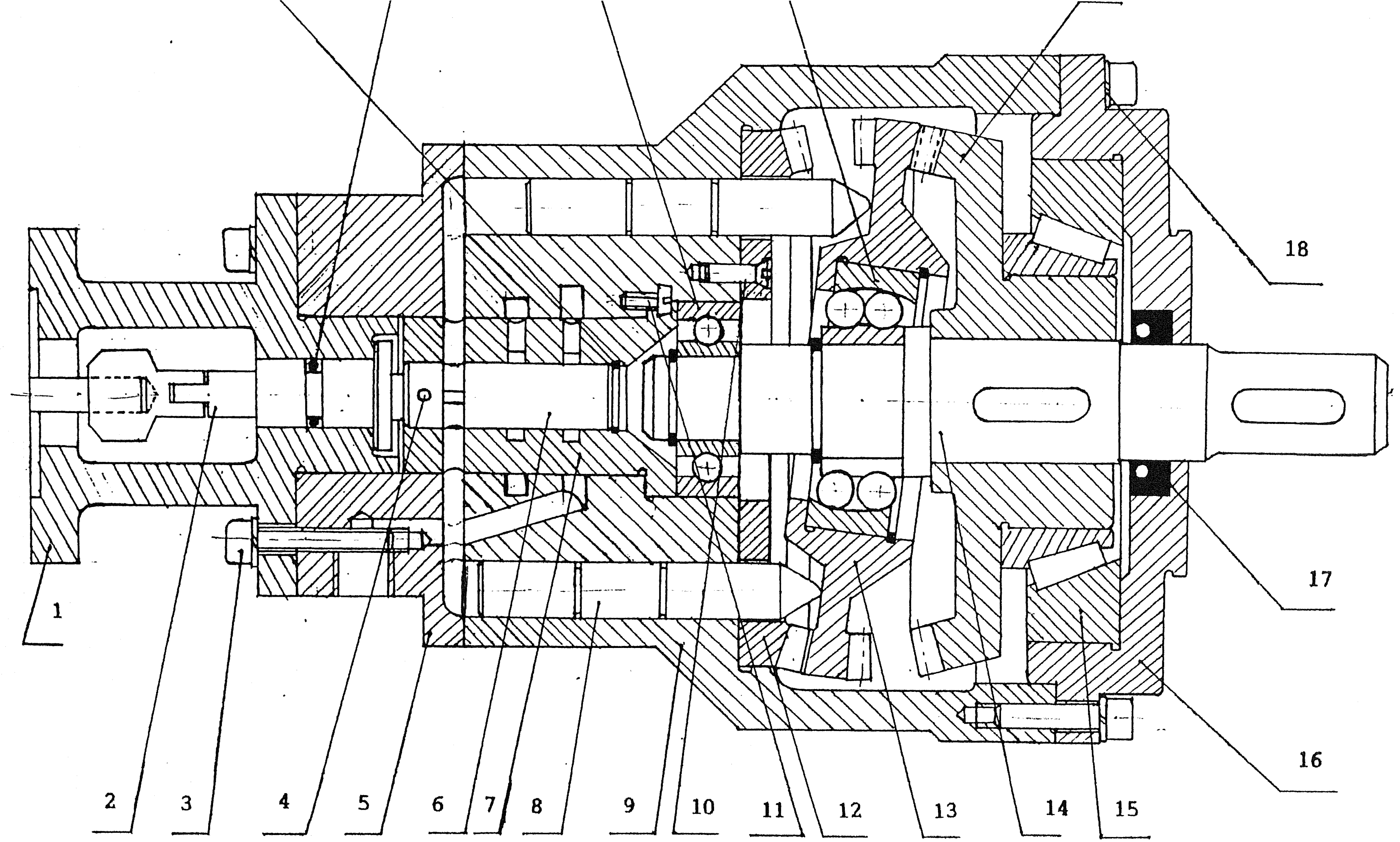


Figure 2.1 Constructional diagram of the proposed EHM - SM.

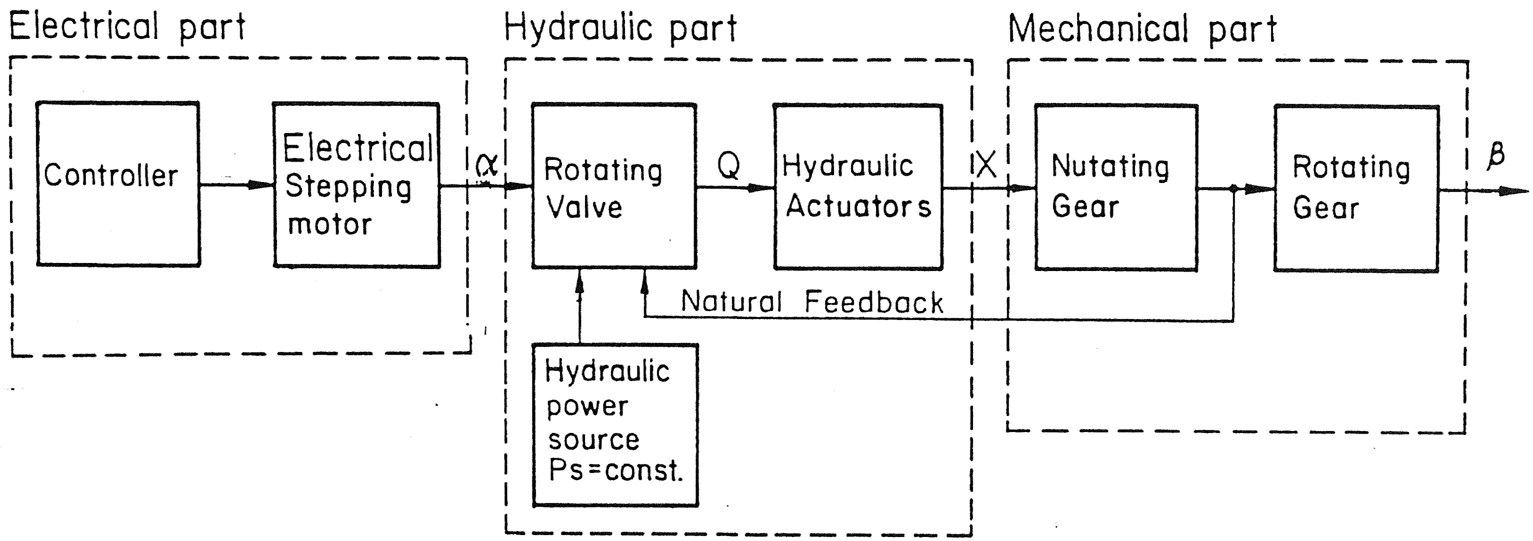


Figure 2.2 Block diagram of the proposed EHM - SM.

of inside passages in the distributor with the help of a lever. The lever can be set either to advance or retreat position. Thus, the direction of output step motion can be reversed and so the proposed motor is bi-directional.

The mechanical portion of the motor consists of :(a) a fixed bevel gear (12) having teeth no. equal to n_R , (b) a nutating bevel gear (13) having same number of teeth as that of the fixed gear and (c) a rotating bevel gear (19). The rotating gear has one tooth more than the nutating gear. The nutating gear is mounted on a self-aligning ball bearing (20) so that it wobbles to maintain continuous meshing with the rotating gear. The output shaft (14) is mounted on two bearings - (i) a single row deep groove ball bearing (21) housed inside the cylinder block at one end, and (ii) a taper roller bearing (15) housed inside the lid (16) at the other end.

2.2 Working Principle Of The Proposed EHM-SM

Let N be the total number of cylinders out of which K are pressurized and (N-K) are depressurized in each step. The resultant axial force will always act in the plane of symmetry of the K pressurized cylinders whether odd or even. For a certain position of the output shaft, say in step (I), cylinders 1,2,...K are being pressurized, then during the next step (i.e step II), cylinders 2,3,...K, K+1 would be pressurized. Now the line of action of the resultant axial force shifts to a new position which lies in the plane of symmetry of cylinders 2,3,...K, K+1. In step III, the pressurization pattern changes : 3,4,...K+1, K+2 are then pressurized and a similar process is repeated. Thus, the resultant axial force rotates by an angle of $\alpha = \frac{2\pi}{N}$ each time.

Now, all the parameters of nutating and rotating gear are the same except that the latter has one tooth more than the former one, and therefore, the tooth thickness of rotating gear is less than that of the nutating one. So, as the resultant force rotates by an angle of α, a corresponding output step is generated on the output shaft.

2.3 Pressurization Pattern Of Cylinders

The hydraulic valve, controlled by an ESM, will create the right pressurization pattern to the cylinders for driving the motor in forward or backward direction. There are two types of regimes in which steps are obtained - (i) full-step and (ii) half-step.

In full-step regime, the proposed motor generates one output step angle equal to β in response to one input step angle equal to α. In this regime, one additional cylinder is pressurized and and one of the pressurized cylinders is depressurized in each step. The pressurization pattern of motor having total number of cylinders equal to N and number of pressurized cylinders equal to K is shown in table 2.1.

In half-step regime, the proposed motor makes its output step angle equal to $\beta/2$ in response to each input step angle equal to $\alpha/2$. In this regime, alternatively one additional cylinder is either pressurized or deactivated. Thus, for a particular output shaft position, if K (1,2,...K) number of cylinders are pressurized , then in the first half-step K+1 (1,2,...K,K+1) cylinders will be pressurized. In the next half step again K cylinders (2,3,...k+1) will be pressurized. The pressurization pattern for N cylinders out of which K are pressurized is shown in table 2.2.

Step	Valve Position	Cylinders Being Pressurized
0	1	1 , 2, ..., K
1	2	2 , 3, ..., K+1
....	....	....
....	....	....
N-K	N-K+1	(N-K+1) , (N-K+2), ..., N
N-K+1	N-K+2	(N-K+2) , (N-K+3), ..., 1
....	....	....
....	....	....
N-1	N	N , 1, ..., K-1
N	1	1 , 2, ..., K

Table 2.1 - Pressurization Pattern For Full - Step Regime .

Step	Valve Position	Cylinders Being Pressurized
0	1	1, 2, ..., K
1	2	1, 2, ..., K+1
2	3	2, 3, ..., K+1
3	4	2, 3, ..., K+1, K+2
....	....	....
....	....	....
2N-2	2N-1	N, 1, ..., K-1
2N-1	2N	N, 1, ..., K-1, K
2N	1	1, 2, ..., K

Table 2.2 - Pressurized Pattern For Half-step Regime.

Pressurization pattern for the first three steps of the proposed motor having fifteen cylinders (N=15) and seven pressurized cylinders (K=7) is shown in figure 2.3.

2.4 Resolution Of EHM - SM.

An angular displacement of output shaft for one input pulse signal is called resolution or step of the motor.

Let n_N, P_N = Number of teeth and circular pitch of nutating gear respectively.

n_R, P_R = Number of teeth and circular pitch of rotating gear respectively.

Then
$$P_N = \frac{2\pi r_1}{n_N} \tag{2.1}$$

and
$$P_R = \frac{2\pi r_1}{n_R} \tag{2.2}$$

where r_1 = average pitch circle radius.

Since $n_R > n_N$, $P_R < P_N$

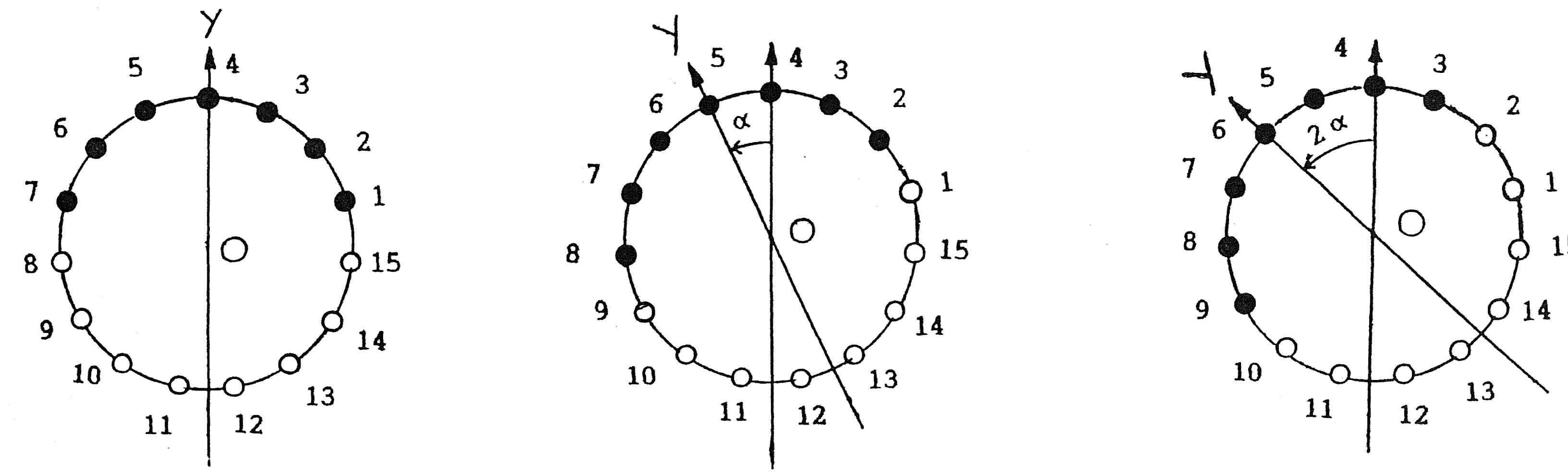
Difference of pitch for a particular pair of teeth,

$$\delta P = \frac{2\pi r_1}{n_N} - \frac{2\pi r_1}{n_R}$$

$$= \frac{2\pi r_1}{n_N n_R} (n_R - n_N) \tag{2.3}$$

In terms of angle, $\delta\beta$ = step angle produced = $\delta P/r_1$ (2.4)

Now, if n_c be the number of pair of teeth in contact between the action of two consecutive resultant forces, then step



a_ Step I

b_ Step II

c_ Step III

● – Pressurized Cylinders

○ – Non-pressurized Cylinders

Figure 2.3 Pressurization pattern of the EHM - SM having N = 15 and K = 7.

produced, $\beta = n_c . \delta\beta$. (2.5)

For full step, $n_c = \dfrac{n_N}{N}$ (2.6)

For half step, $n = \dfrac{n_N}{2N}$ (2.7)

where, N = Number of total cylinders.

The step angle for full step and half step regime are given by :

$$\beta = \frac{n_N}{N} \times \frac{2\pi r_1}{n_N \, n_R} \, (n_R - n_N) \, \frac{1}{r_1}$$

$$= \frac{2\pi}{Nn_R} \, (n_R - n_N)$$

$$= \frac{360^0}{Nn_R} \, (n_R - n_N) \qquad (2.8)$$

$$\beta_H = \frac{n_N}{2N} \times \frac{2\pi r_1}{n_N \, n_R} \, (n_R - n_N) \, \frac{1}{r_1}$$

$$= \frac{180^0}{Nn_R} \, (n_R - n_N) \qquad (2.9)$$

For high resolution, $(n_R - n_N)$ should be minimum and N should be as large as possible. Possible minimum value of $(n_R - n_N)$ is one. For selecting the total number of cylinders leakage between two consecutive cylinders should be taken into account. When the cylinders are arranged in a circular array, the center distance between two consecutive cylinders should not be less than 2.3d, where d is the diameter of cylinder or piston.

Taking above considerations into account, for the proposed EHM-SM,

$$n_R = 49, \; n_N = 48 \text{ and } N = 15$$

Thus, the resolution of the proposed motor for full step and half step regimes are given by :

$$\beta = 0.489^0 \quad \text{and} \quad \beta_H = 0.2445^0$$

2.5 Hydraulic Valve

Hydraulic valve in the proposed motor,as discussed earlier, provides a predetermined pressurization pattern. As the pressurization pattern changes steps are generated. Two types of valves are used for generation of full-step and half-step.

Type-I valve used for full-step is characterized by :

$$\varphi_1 = K\alpha \quad \text{where, } \varphi_1 = \text{Valve land angle} \tag{2.10}$$

where K = Number of pressurized cylinders in one step.

$$\alpha = \text{Input step angle} = \frac{360^0}{N} \tag{2.11}$$

Cross-sectional area of the valve (type I) is shown in Figure 2.4. Two possible initial positions of the valve land are indicated.

For the first initial position (a) , as the valve rotates by an angle of $\alpha/2$, pressurization pattern remains unchanged. However, for next $\alpha/2$ angular rotation of the valve, pressurization pattern changes from 1,2,....K to 2,3,....K+1. So, resultant force gets shifted by an angle of α and one step is generated.

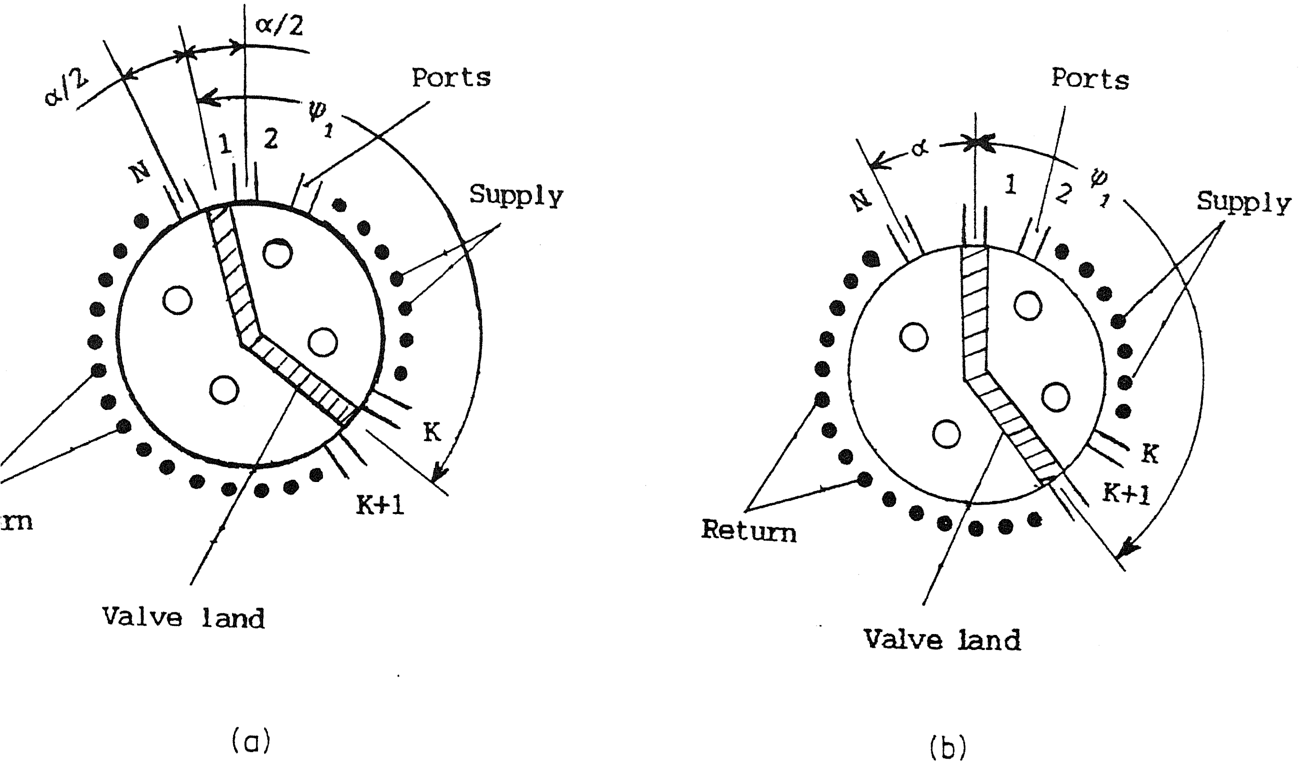
For the second initial position (b), as the valve rotates by an angle of $\alpha/2$, the pressurization pattern changes but for the next half input step , there is no change in pressurization pattern.

So, in both cases, for a particular input angle α, only one output step is generated. So the motor works in the full-step regime only for type-I valve.

Type-II valve used for half-step regime is characterized by :

$$\varphi_1 = (2K+1)\frac{\alpha}{2} \tag{2.12}$$

Cross-sectional area of the valve (type II) at land is shown



(a) (b)

Figure 2.4 Cross-sectional area of type - I valve ($\psi_1 = K\alpha$).

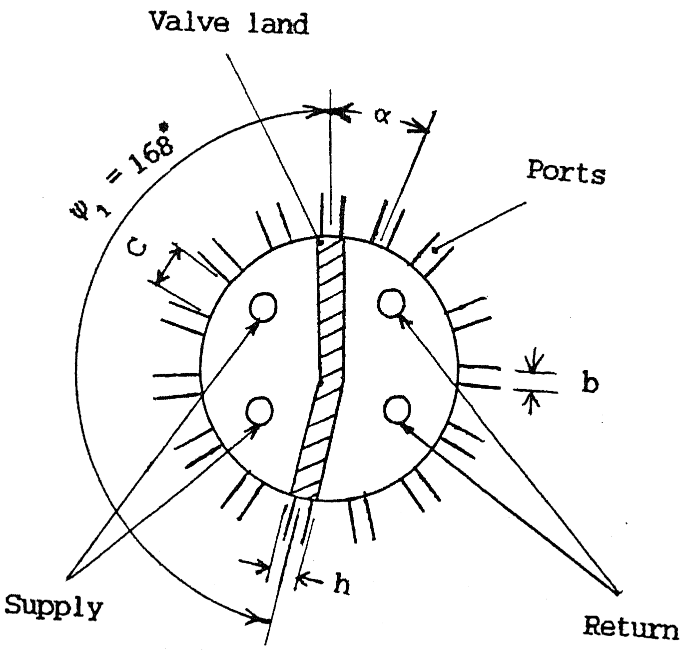


Figure 2.5 Cross-sectional area of full-step valve used in the motor ($\psi_1 = 168^0$).

in Figure 2.6. For the particular initial position shown, as the valve rotates by an angle of $\alpha/2$, pressurization pattern changes from 1,2,...K to 1,2,...,K,K+1. For the next input half-step angle ($\alpha/2$), again the pressurization pattern changes and this time from 1,2,....K+1 to 2,3,....,K+1.

So, for a particular input step angle α , two steps are generated. The resultant force shifts by an angle equal to $\alpha/2$ each time instead of α. Thus the steps generated are the half steps and the motor works in the half-step regime.

For the proposed motor, optimum number (i.e torque produced is maximum) of cylinders pressurized is , $k = 7$ as $k = k^* = \frac{k \pm 1}{2}$ and $\alpha = 24^0$

Hence, for full-step, $\varphi_1 = 168^0$

and for half-step, $\varphi_1 = 180^0$

These two cases are shown in Figure 2.5 and Figure 2.7 respectively.

2.6 Fluid Flow Required by EHM - SM

Average theoretical fluid flow rate is defined as fluid flow rate per unit of time without taking leakage into account, whereas, instantaneous fluid flow rate is the fluid flow rate defined at every instant of time .

2.6.1 Average Theoretical Flow Rate required by EHM - SM working in Full - Step regime.

Let N be the total number of cylinders and K be the number of pressurized cylinders.

If only one cylinder is pressurized , the flow rate required by the motor is given by

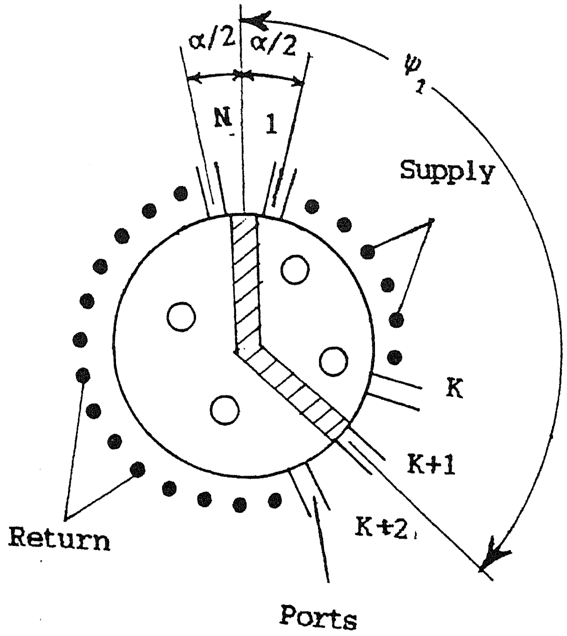


Figure 2.6 Cross-sectional area of type - II valve

$[\ \psi_1 = (2K + 1)\frac{\alpha}{2}\ ]$.

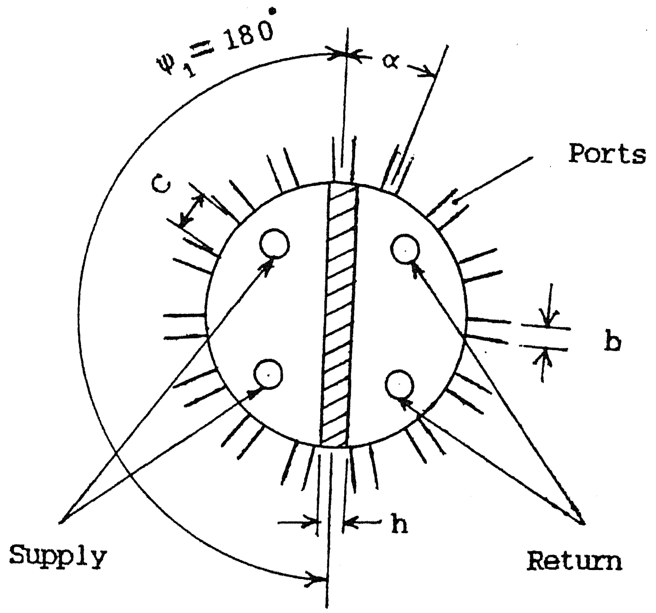


Figure 2.7 Cross-sectional area of half-step valve used in the motor ($\psi_1 = 180^0$).

$$q_1^N = A \cdot S_1^N \ cm^3 \qquad (2.13)$$

where A is the cross-sectional area (cm^2) of piston and S_1^N is the effective axial travel of the piston.

If K cylinders are pressurized , the fluid flow required by the motor in one step is given by

$$q_K^N = A \cdot S_K^N \ cm^3 \qquad (2.14)$$

Average theoretical flow rate , denoted by Q_K^N , required by the motor is given by

$$Q_K^N = A \cdot S_K^N \cdot f \ \ (cm^3/ sec.) \qquad (2.15)$$

where f = stepping frequency of motor.

The effective axial travel (S_K^N) can be found out based on adopted pressurization pattern and number of cylinders (K) being pressurized [19]

$$S_K^N = 2\ r \tan (2\gamma) \sin (\alpha/2) \sin (K \alpha /2) \qquad (2.16)$$

where r = radius of pitch circle locating the cylinders , cm.

γ = tilt angle , deg.

Putting equation (2.16) into equation (2.15) , we get ,

$$Q_K^N = 2\ A\ r\ f \tan (2\gamma) \sin (\alpha /2) \sin (K \alpha /2) \qquad (2.17)$$

Thus, average theoretical flow rate required by the motor depends on area of piston (A) , radius of pitch circle of cylinders (r), stepping frequency (f), tilt angle (γ), number of cylinders pressurized (K) and total number of cylinders (N).

The variation of average theoretical flow rate with stepping frequency is shown in Figure 2.8 for N = 15, r= 3.4 cm., d = 1 cm. and $\gamma = 7^0$ which are the proposed dimensions of the motor. The characteristics are linear in nature and as the stepping frequency (f) increases, the average theoretical flow rate required by the motor also increases.

Assuming B = A.r.ω tan (2γ) and $\bar{Q}_K^N$ = non-dimensional avg.

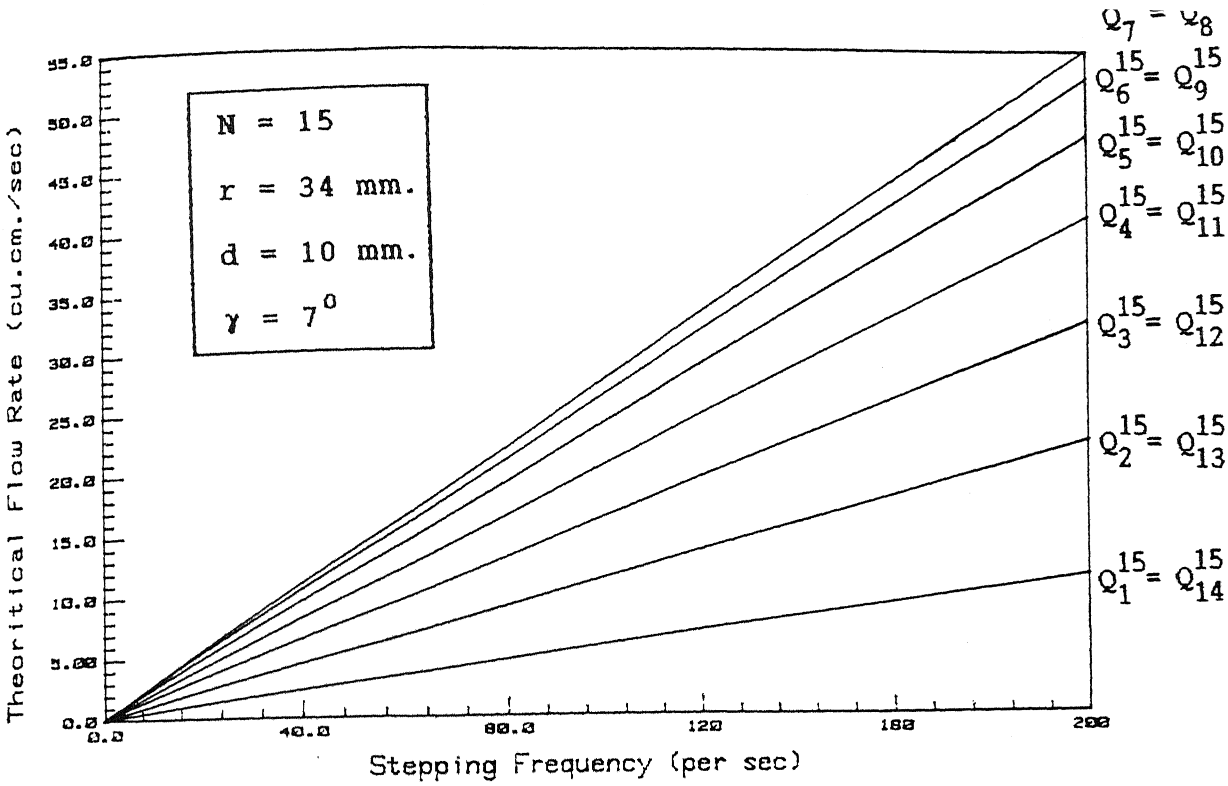


Figure 2.8 Variation of average theoretical flow rate required by EHM -SM with stepping frequency.

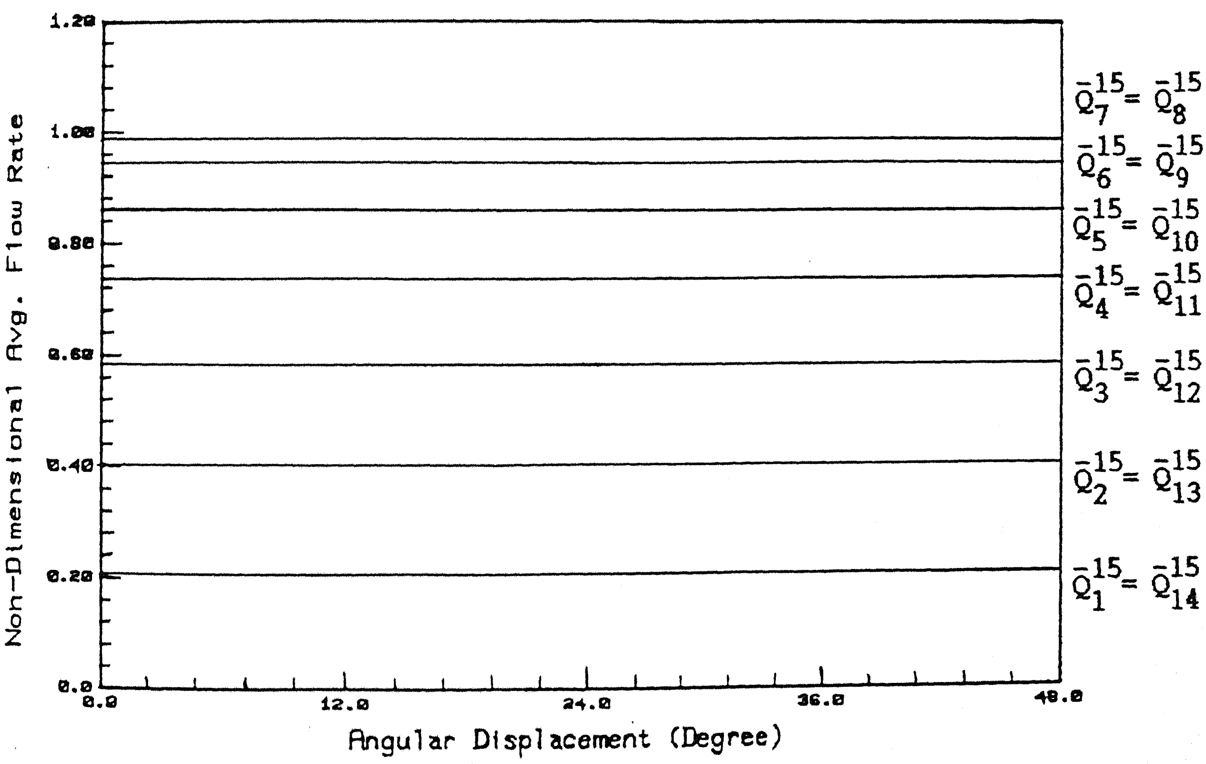


Figure 2.9 Non - dimensional average theoretical flow rate required by the EHM - SM working in full-step regime (N = 15)

theoretical flow rate = Q_K^N /B, the equation (2.17) becomes

$$\bar{Q}_K^N = \frac{N}{\pi} \text{Sin } (\alpha/2).\text{Sin } (K\, \alpha/2) \qquad (2.18)$$

For the proposed motor having N = 15, equation (2.18) becomes :

$$\bar{Q}_K^{15} = \frac{15}{\pi} \text{Sin } (12^0) \text{ Sin } (K\ 12^0) \qquad (2.19)$$

Figure 2.9 shows graphical representation of equation (2.19) for $1 \le K \le 14$ from where the following observations can be made:

1) When no cylinder (K = 0) or all cylinders (K = N) are pressurized, no flow rate is required, i.e

$$\bar{Q}_0^{\ N} = \bar{Q}_N^{\ N} = 0$$

2) Maximum flow rate required occurs when $K = K^* = \frac{N \pm 1}{2}$

3) The closer K is to K^*, the lesser is the difference in flow rate when pressurization pattern changes from K to K+1.

2.6.2 Non - Dimensional Instantaneous Flow Rate required by the EHM - SM working in Full - Step regime

Instantaneous flow rate depends on the velocity of pistons reciprocating inside the cylinders. For one particular cylinder , the instantaneous flow rate is given by :

$$q_\Phi = A \cdot V_\Phi \qquad (2.20)$$

If the proposed motor has N number of cylinders out of which K cylinders are pressurized , then instantaneous flow rate is given by :

$$Q_K^{N\Phi} = A \cdot V_{\Phi_1} + A \cdot V_{\Phi_2} + \ldots\ldots\ldots + A \cdot V_{\Phi_k} \qquad (2.21)$$

The axial travel of piston as the nutating gear rotates by an angle ϕ from its initial is given by Fig. 2.10 :

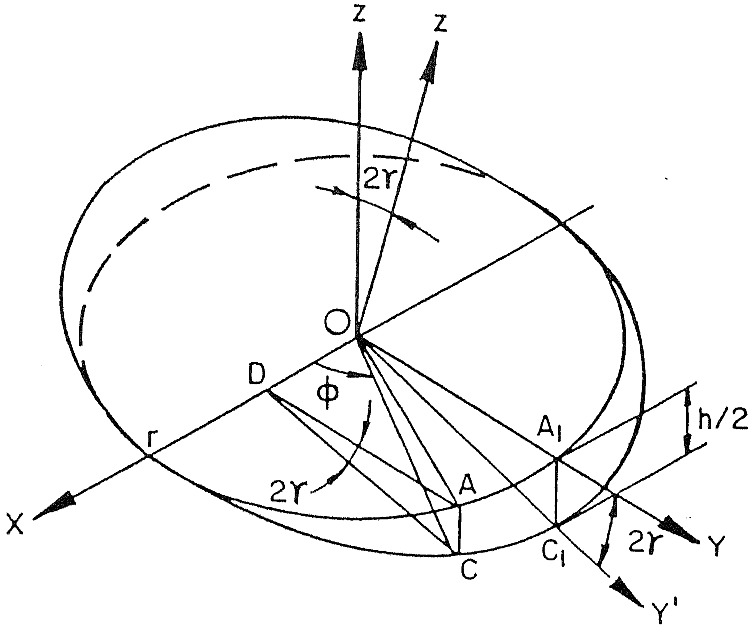


Figure 2.10 Kinematics of Piston

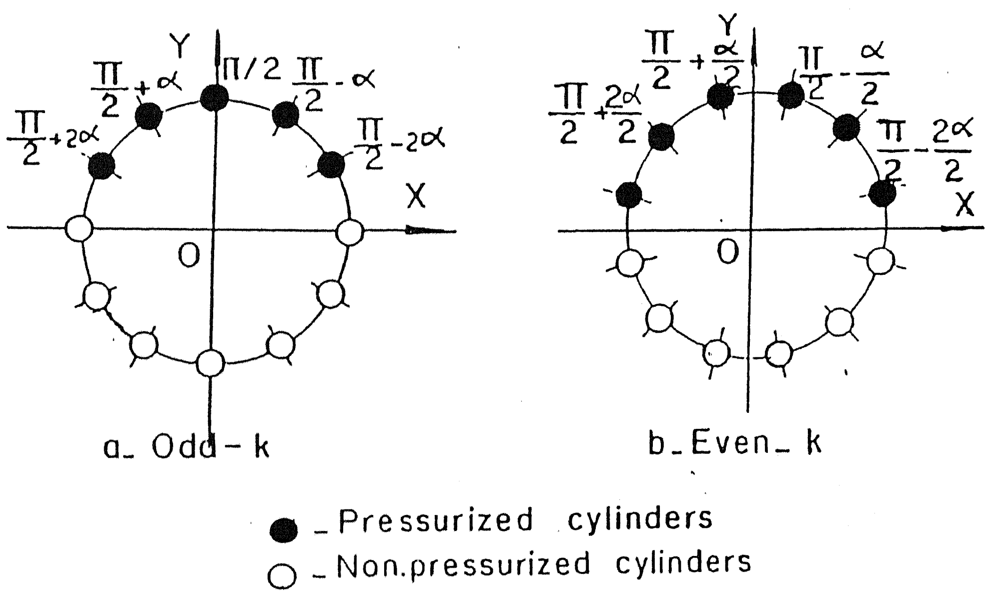


Figure 2.11 General pressurization pattern for odd - K and even - K

$$Z_\phi = AC = CD \tan(2\gamma)$$

$$= OC \sin\phi \tan(2\gamma)$$

$$\text{or} \quad Z_\phi = r \sin\phi \tan(2\gamma) \tag{2.22}$$

$$V_\phi = \frac{dZ_\phi}{dt} = r \tan(2\gamma) \cos\phi \frac{d\phi}{dt}$$

$$= r\,\omega \tan(2\gamma) \cos\phi$$

$$\text{or } V_\phi = \frac{B \cos\phi}{A} \tag{2.23}$$

Substituting V_ϕ in equation (2.21), we get,

$$Q_K^{N\Phi} = B \sum_{i=1}^{k} \cos\phi_i \tag{2.24}$$

where ϕ_i defines the instantaneous position of i-th piston.

General pressurization patterns for odd-k and even-k are shown in Fig. 2.11 (a) and (b) respectively.

$$\text{For odd-k,} \sum_{i=1}^{k} \cos\phi_i = \sin(\alpha - \phi) \left[1 + 2 \sum_{i=1}^{(k-1)/2} \cos(i\alpha) \right] \tag{2.25}$$

$$\text{For even-k,} \sum_{i=1}^{k} \cos\phi_i = 2 \sin(\alpha - \phi) \sum_{i=1}^{(k-1)/2} \cos(2i - 1)\frac{\alpha}{2} \tag{2.26}$$

$$\text{Then}, \quad Q_{KO}^{N\Phi} = B \sin(\alpha - \phi) \left[1 + 2 \sum_{i=1}^{(k-1)/2} \cos(i\alpha) \right]$$

$$\bar{Q}_{KO}^{N\Phi} = \frac{Q_{KO}^{N\Phi}}{B} = \sin(\alpha - \phi) \left[1 + 2 \sum_{i=1}^{(k-1)/2} \cos(i\alpha) \right] \tag{2.27}$$

$$\text{and} \quad Q_{Ke}^{N\Phi} = 2 B \sin(\alpha - \phi) \sum_{i=1}^{k/2} \cos(2i - 1)\frac{\alpha}{2}$$

$$\bar{Q}_{Ke}^{N\Phi} = \frac{Q_{Ke}^{N\Phi}}{B} = 2 \sin(\alpha - \phi) \sum_{i=1}^{k/2} \cos(2i - 1)\frac{\alpha}{2} \tag{2.28}$$

Fig.2.12 shows graphical representation of equations (2.27) and (2.28) for N = 15, from which following observations can be

made:

1) Maximum instantaneous flow rate required by the motor occurs when optimum number of cylinders are pressurized,

$$\text{i.e} \quad K = K^{*} = \frac{N \pm 1}{2}$$

2) When $1 \le K \le K^{*}$, then $Q_{K}^{N\Phi} < Q_{K+1}^{N\Phi}$

and when $K^{*} \le K \le N - 1$, then $Q_{k+1}^{N\Phi} < Q_{k}^{N\Phi}$

2.6.3 Average Theoretical Flow Rate required by the EHM - SM working in Half- Step regime.

In case of half- step regime, the pressurization pattern changes from K to K+1 in the first half-step and from K+1 to K in the second half-step . If the effective axial travel of pistons in the first and second half-steps are denoted by S_{k+1}^{NH} and S_{k}^{NH} respectively, then flow required by the motor during one step (comprising of two half-steps) is given by :

$$q_{K}^{NH} + q_{K+1}^{NH} = A \left(S_{K+1}^{NH} + S_{K}^{NH} \right) \quad cm^{3} \qquad (2.29)$$

So, Avg. theoretical flow rate required by the motor is given by

$$Q_{K}^{NH} = A \left(S_{K+1}^{NH} + S_{K}^{NH} \right) f \quad cm^{3}/sec \qquad (2.30)$$

Now , S_{k}^{NH} is given by [19] :

$$S_{K}^{NH} = 2\, r \tan(2\gamma) \operatorname{Sin}(\alpha/4) \sum_{i=1}^{k} (-1)^{k-i} \operatorname{Sin}(2i-1).\frac{\alpha}{4} \qquad (2.31)$$

$$\text{So, } S_{K+1}^{NH} = 2\, r \tan(2\gamma) \operatorname{Sin}(\alpha/4) \sum_{i=1}^{k+1} (-1)^{k+1-i} \operatorname{Sin}(2i-1).\frac{\alpha}{4} \qquad (2.32)$$

Substituting equation (2.31) and (2.32) in equation (2.30), we get,

$$Q_{K}^{NH} = 2\, A\, r\, f \tan(2\gamma) \operatorname{Sin}(\alpha/4) \left[\sum_{i=1}^{k} (-1)^{k-i} \operatorname{Sin}(2i-1).\frac{\alpha}{4} \right.$$

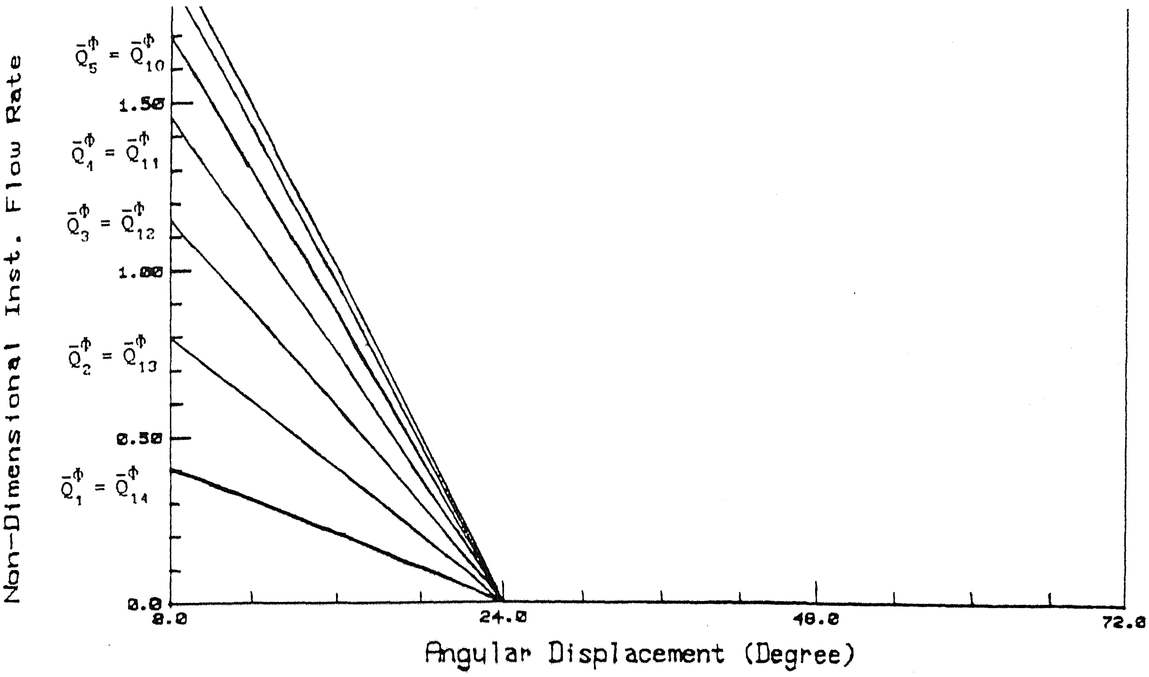


Figure 2.12 Non - dimensional instantaneous flow rate required by the EHM - SM working in full-step regime (N = 15)

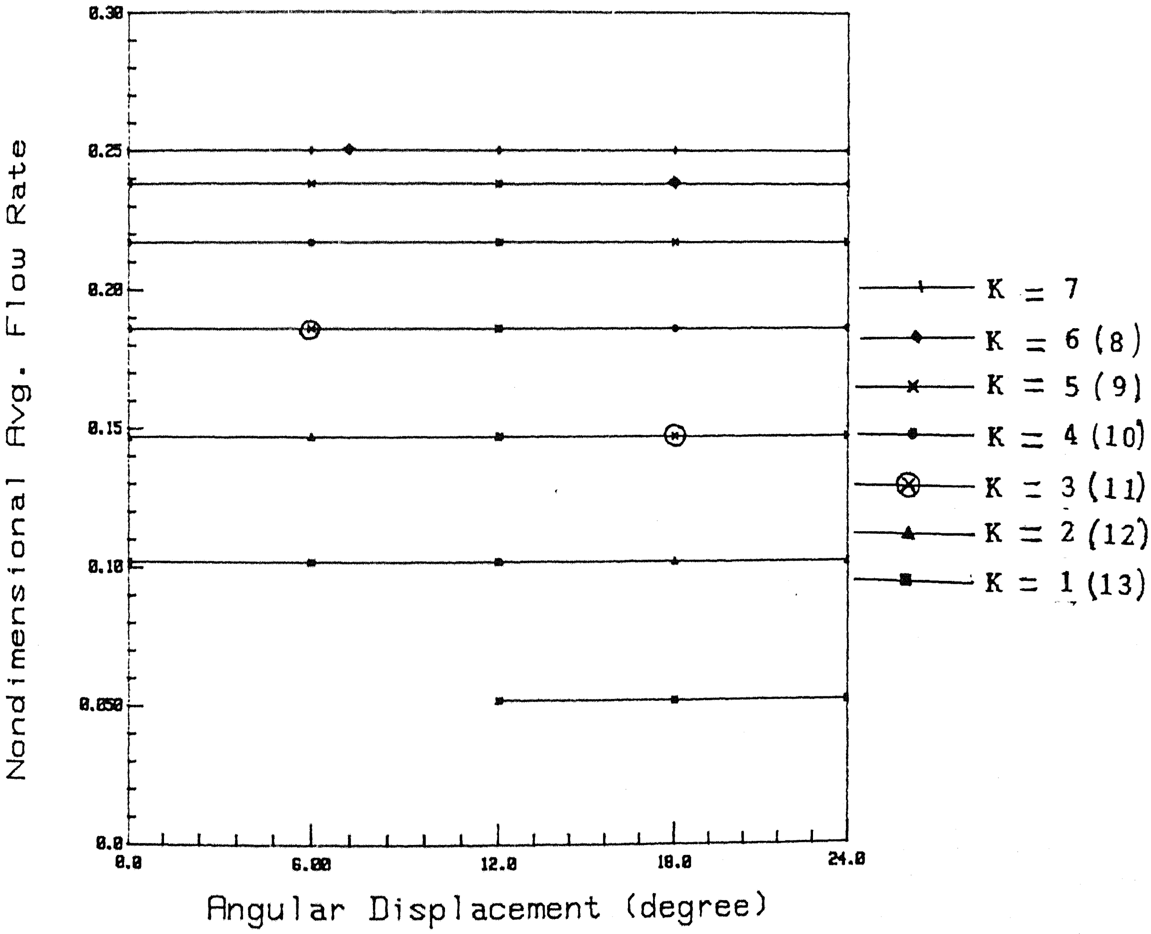


Figure 2.13 Non-dimensional Average theoretical flow rate required by the EHM - SM working in the half-step regime (N = 15)

$$+ \sum_{i=1}^{k+1} (-1)^{k+1-i} \operatorname{Sin} (2i-1).\frac{\alpha}{4} \Bigg]$$

$$= \frac{N\ A\ r\ \omega \tan (2\gamma) \operatorname{Sin} (\alpha/4)}{\pi} \Bigg[\sum_{i=1}^{k} (-1)^{k-i} \operatorname{Sin} (2i-1).\frac{\alpha}{4}$$

$$+ \sum_{i=1}^{k+1} (-1)^{k+1-i} \operatorname{Sin} (2i-1).\frac{\alpha}{4} \Bigg]$$

$$= \frac{B\ N \operatorname{Sin} (\alpha/4)}{\pi} \Bigg[\sum_{i=1}^{k} (-1)^{k-i} \operatorname{Sin} (2i-1).\frac{\alpha}{4}$$

$$+ \sum_{i=1}^{k+1} (-1)^{k+1-i} \operatorname{Sin} (2i-1).\frac{\alpha}{4} \Bigg] \qquad (2.33)$$

$$\bar{Q}_k^{NH} = \frac{N \operatorname{Sin} (\alpha/4)}{\pi} \Bigg[\sum_{i=1}^{k} (-1)^{k-i} \operatorname{Sin} (2i-1).\frac{\alpha}{4}$$

$$+ \sum_{i=1}^{k+1} (-1)^{k+1-i} \operatorname{Sin} (2i-1).\frac{\alpha}{4} \Bigg] \qquad (2.34)$$

Figure 2.13 shows graphical representation of equation (2.34) for N = 15. from where the following observations can be made:

1) The motor requires more flow rate during first half-step than the second one when K is not equal to K^*.

2) To produce a uniform flow rate , optimum number of cylinders are pressurized.

2.6.4 Non - dimensional Instantaneous Flow Rate required by EHM -SM working in half- step regime.

Similar to the case of full-step regime, the equations (2.24),(2.25) and (2.26) also hold good for half-step regime.

CASE-I

When K is odd , then K+1 is even.

For the first half- step,

$$Q_{K+1}^{NH\Phi} = 2\ B\ \mathrm{Sin}\ (\alpha/2 - \phi) \sum_{i=1}^{(k+1)/2} \mathrm{Cos}\ (2i - 1)\frac{\alpha}{2} \qquad (2.35)$$

And for the second half-step,

$$Q_{K}^{NH\Phi} = B\ \mathrm{Sin}\ (\alpha/2 - \phi) \left[1 + 2 \sum_{i=1}^{(k-1)/2} \mathrm{Cos}\ (i\alpha) \right] \qquad (2.36)$$

CASE-II

When K is even , K+1 is odd.

For the first half- step,

$$Q_{K+1}^{NH\Phi} = B\ \mathrm{Sin}\ (\alpha/2 - \phi) \left[1 + 2 \sum_{i=1}^{K/2} \mathrm{Cos}\ (i\alpha) \right] \qquad (2.37)$$

And for the second half-step,

$$Q_{K}^{NH\Phi} = 2\ B\ \mathrm{Sin}\ (\alpha/2 - \phi) \sum_{i=1}^{K/2} \mathrm{Cos}\ (2i - 1)\frac{\alpha}{2} \qquad (2.38)$$

Fig. 2.14 shows the graphical representation of the above four equations from where the following observations can be made :

1) As the pressurization pattern changes from first half-step to second half-step , the flow rate required by the motor also changes during these two half-steps.

2) The pattern of variation of flow rate during a half-step remains the same.

2.7.1 Theoretical Torque Developed by the EHM - SM working in the Full - Step Regime.

The forces acting on the nutating bevel gear is shown in Fig. 2.15. Normal force (G_n) at the point of contact of the piston can be resolved into two components - horizontal (G) and vertical (G_v).

For one piston being pressurized , the axial force is given by :

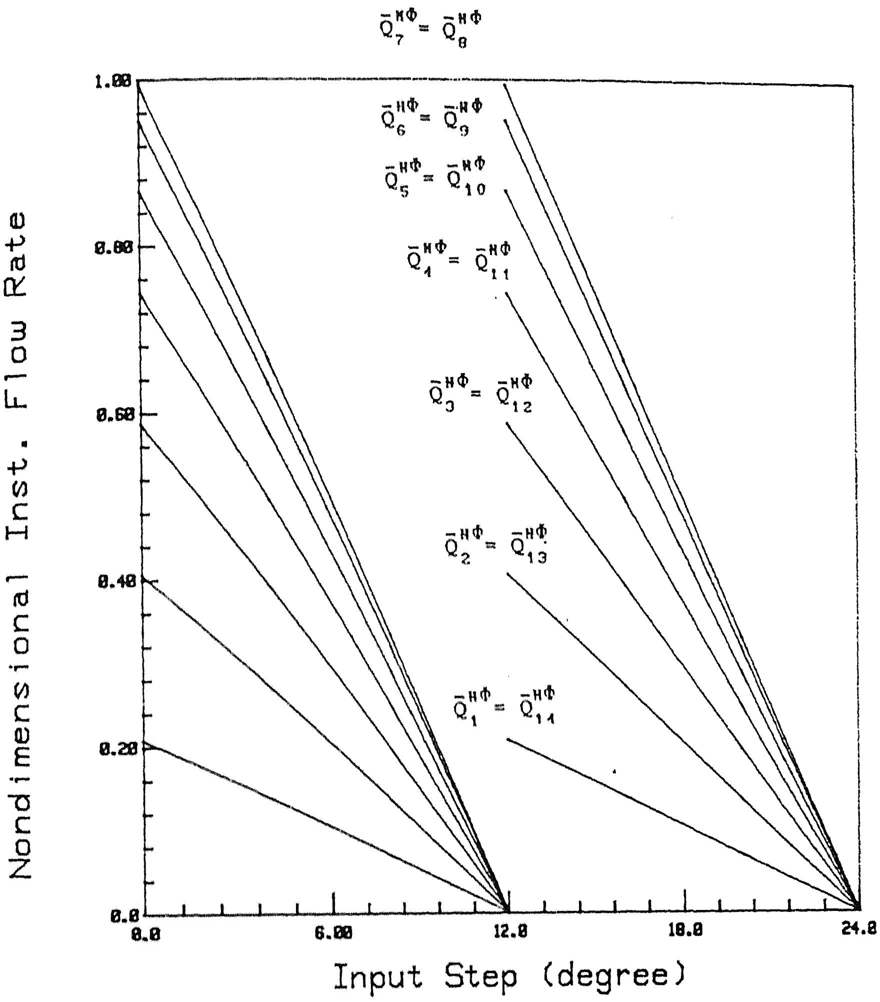
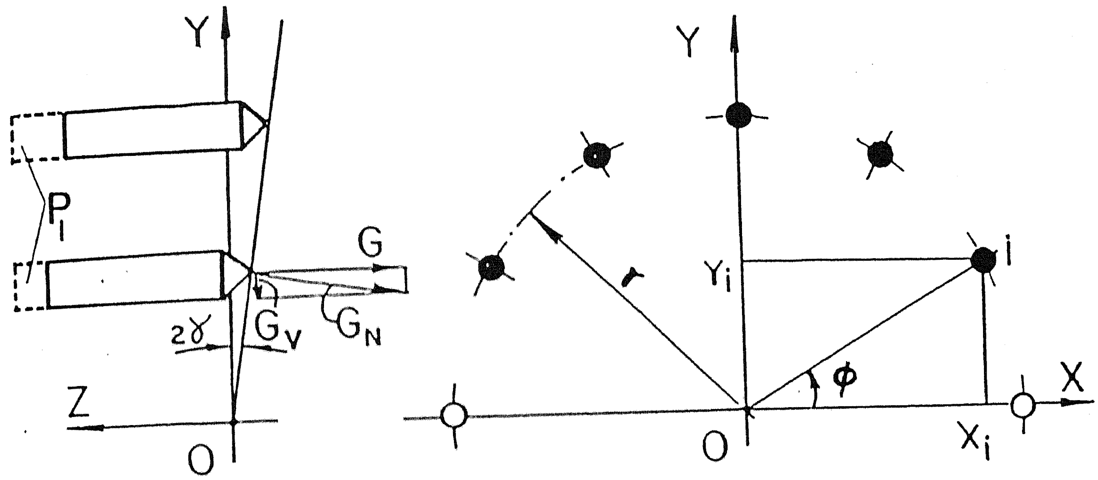


Figure 2.14 Non-dimensional instantaneous flow rate required by the EHM - SM working in the half-step regime (N = 15)



Pressurized Cylinders

Non-pressurized Cylinders

Figure 2.15 Forces acting on the nutating bevel gear by pressurized pistons

$$G = A\,(P_1 - P_0) \tag{2.39}$$

where A is the area of piston , P_1 is the pressure in the forward chamber and P_0 is the pressure in the backward chamber.

In general , for K cylinders being pressurized , the axial force is given by :

$$G_K^N = \sum_{i=1}^{K} G_i = K\,G \tag{2.40}$$

Moments caused by pressurized pistons around OX and OY are given by :

$$M_K^{NX} = \sum_{i=1}^{K} G\, Y_i^{\Phi} = G \sum_{i=1}^{K} Y_i^{\Phi} \tag{2.41}$$

and $$M_K^{NY} = \sum_{i=1}^{K} G\, X_i^{\Phi} = G \sum_{i=1}^{K} X_i^{\Phi} \tag{2.42}$$

For the equilibrium or end-of-step position as shown in Fig. 2.16, due to the symmetry of pressurization pattern ,

$$M_K^{NX} = M_{K\,max}^{NX} \quad \& \quad M_K^{NY} = 0 \tag{2.43}$$

The relation between instantaneous ($0 < \phi < \frac{2\pi}{N}$) and end- of- step ($\phi = 0$ or $\frac{2\pi}{N}$) co-ordinates are given by [20] :

$$[X_i^{\Phi}\ Y_i^{\Phi}\ Z_i^{\Phi}\ ]^T = R^T(z,\Phi)\ [X_i\ Y_i\ Z_i\ ]^T \tag{2.44}$$

The rotational matrix is given by :

$$R^T(z,\Phi) = \begin{bmatrix} \cos\Phi & \sin\Phi & 0 \\ -\sin\Phi & \cos\Phi & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{2.45}$$

It should be noted that only moment about OX i.e M_K^{NX} will contribute in developing output torque.

$$\text{So, } M_K^{NX} = M_K^N \tag{2.46}$$

Substituting equations (2.44) , (2.45) & (2.46) into (2.41) we get ,

$$M_K^N = G\ [-\sin\Phi\ \ \cos\Phi\ ] \sum_{i=1}^{K} [\ X_i\ \ Y_i\ ]^T \tag{2.47}$$

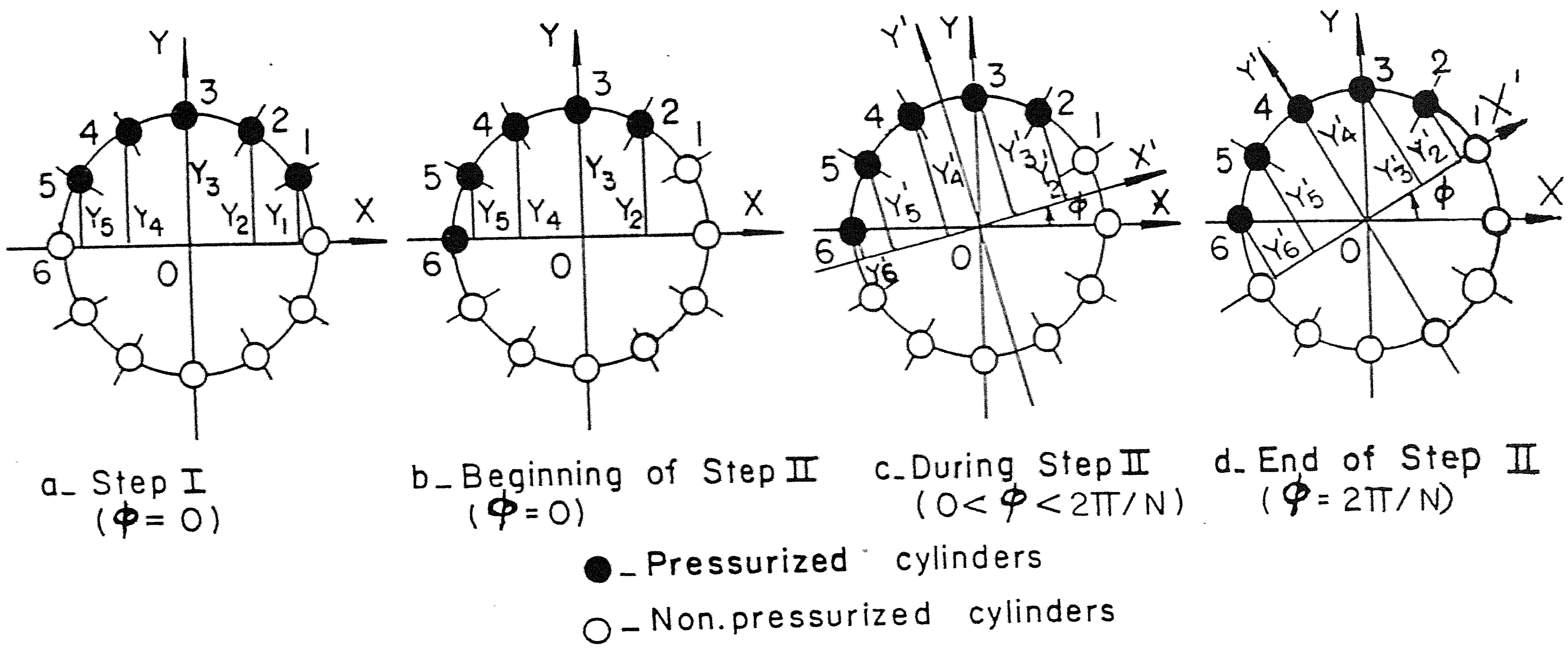


Figure 2.16 Transition of step - I to step - II

Now, consider the forces transmitted from the nutating gear to the rotating gear as shown in Fig. 2.17.

F = Resultant force.

F_T = Tangential force.

F_r = Radial component.

F_a = Axial component.

$\bar{\gamma}$ = Pressure angle.

θ = Pitch-cone angle.

$\gamma = 90^0 - \theta$ = tilt angle.

Force components are interrelated by :

$$F_T = \frac{T_K^N}{r_2} \qquad (2.48)$$

$$F_r = F_T \tan\bar{\gamma} \cos\theta \qquad (2.49)$$

and $$F_a = F_T \tan\bar{\gamma} \sin\theta \qquad (2.50)$$

where, T_K^N = Output torque

r_2 = Mean radius of rotating gear.

Now , $M_K^N = F^{*} . OC$

$$= \frac{F_a}{\cos\gamma} . \frac{r_2}{\cos\gamma} = \frac{F_T \tan\bar{\gamma} \sin\theta}{\cos\gamma} . \frac{r_2}{\cos\gamma}$$

Since $\cos\gamma = \sin\theta$, the expression for the moment M_K^N becomes:

$$M_K^N = \frac{T_K^N \tan\bar{\gamma}}{\cos\gamma}$$

So , $$T_k^N = \frac{M_K^N \cos\gamma}{\tan\bar{\gamma}}$$

$$= \frac{G \cos\gamma}{\tan\bar{\gamma}} [- \sin\Phi \quad \cos\Phi] \sum_{i=1}^{K} [X_i \quad Y_i]^T \qquad (2.51)$$

Consider the particular case when only one cylinder is

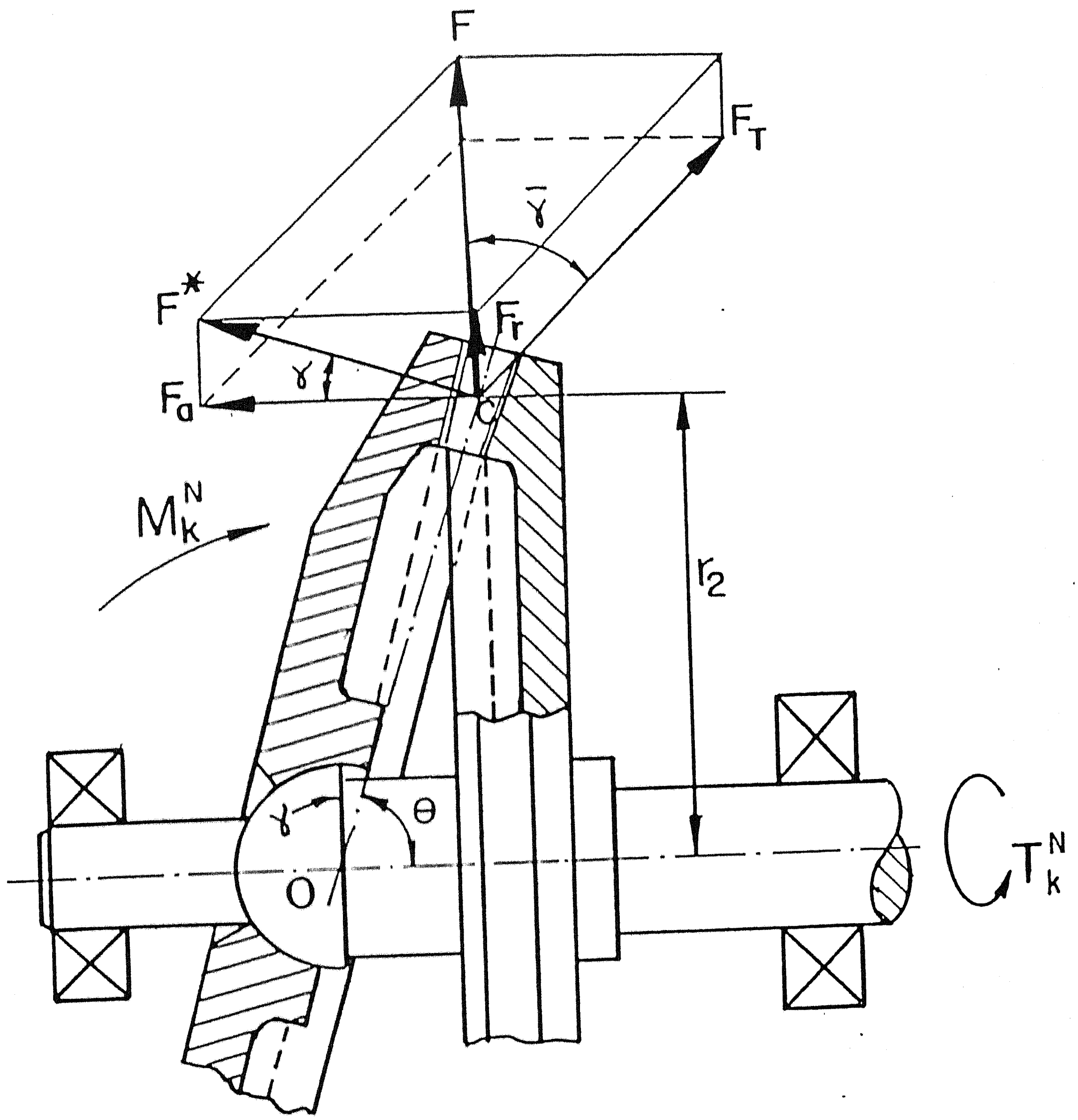


Figure 2.17 Forces transmitted from the nutating gear to the rotating gear

pressurized out of N cylinders as shown in Fig. 2.18.

Then $X_1 = -r \, \mathrm{Sin}\alpha$

$Y_1 = r \, \mathrm{Cos}\alpha$

Substituting these values in equation (2.51) , we get ,

$$T_1^N = \frac{G \cos\gamma}{\tan\bar{\gamma}} \left[-\,\mathrm{Sin}\Phi \quad \mathrm{Cos}\Phi \right] \begin{bmatrix} -\,r \, \mathrm{Sin}\alpha \\ r \, \mathrm{Cos}\alpha \end{bmatrix}$$

$$= \frac{r \, G \cos\gamma}{\tan\bar{\gamma}} \, \mathrm{Cos} \, (\alpha - \phi) \tag{2.52}$$

$$T_1^N \Big|_{min} = T_1^N \Big|_{\Phi = 0} = \frac{r \, G \cos\gamma}{\tan\bar{\gamma}} . \cos\alpha \tag{2.53}$$

$$T_1^N \Big|_{max} = T_1^N \Big|_{\Phi = \alpha} = \frac{r \, G \cos\gamma}{\tan\bar{\gamma}} \tag{2.54}$$

Thus , at the start-of-step , instantaneous torque is minimum and at the end-of-step , the instantaneous torque is maximum which is also known as the holding torque or stall torque .

For the two cylinders being pressurized , following the same way , we get ,

$$T_2^N = 2 \, T_1^N \Big|_{max} \cos \, (\alpha - \phi) \tag{2.55}$$

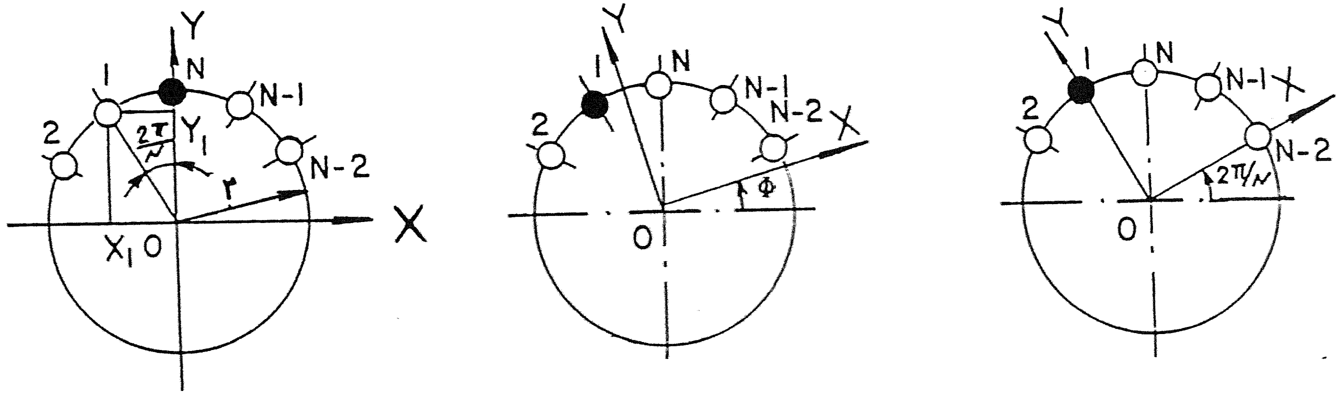
In general , T_K^N can be expressed as :

$$T_K^N = T_1^N \Big|_{max} C_K^N \cos \, (\alpha - \phi) \tag{2.56}$$

Holding torque is given by : $T_K^N \Big|_{\Phi=\alpha} = T_1^N \Big|_{max} . \, C_K^N$ (2.57)

$$\text{where} \quad C_K^N = \frac{\sum_{i=1}^{K} Y_i}{r} = 1 + 2 \sum_{i=1}^{(k-1)/2} \cos \, (i\alpha) \tag{2.58}$$

when K is odd .



(a) Initial step (b) Intermediate position (c) End of step

$\phi = 0$ $0 < \phi < \frac{2\pi}{N}$ $\phi = \frac{2\pi}{N}$

● – Pressurized Cylinders

○ – Non-pressurized Cylinders

Figure 2.18 Pressurization pattern when only one cylinder is pressurized

and $$C_K^N = 2 \sum_{i=1}^{K/2} \cos\ (2i - 1)\frac{\alpha}{2} \qquad (2.59)$$

when K is even .

Equation (2.56) can be expressed in different form as:

$$\frac{T_K^N}{T_1^N\Big|_{max}} = \bar{T}_K^N = C_K^N \cos\ (\ \alpha - \phi\ ) \qquad (2.60)$$

where $\bar{T}_K^N$ is the non-dimensional instantaneous torque developed by EHM-SM in the full-step regime. Fig. 2.19 shows the graphical representation of equation (2.60) when N = 15 and $1 \le K \le 14$ from where the following observations can be made :

1) When the motor is at rest (i.e $\phi = 0$) , output torque is maximum which is known as the holding torque. When switching to next step occurs , torque drops to its minimum value and then as ϕ increases , the torque also increases gradually to its maximum value at the end of step ($\phi = \frac{2\pi}{N}$).

2) When $K < K^*$, then $T_K^N < T_{K+1}^N$

and when $K > K^*$, then $T_{K+1}^N > T_K^N$

3) When no cylinder is pressurized (K = 0) or all cylinders are pressurized (K = N) , then torque developed is zero . This is because of the fact that the effective radius of the resultant force becomes zero.

Coefficient of torque pulsation is given by :

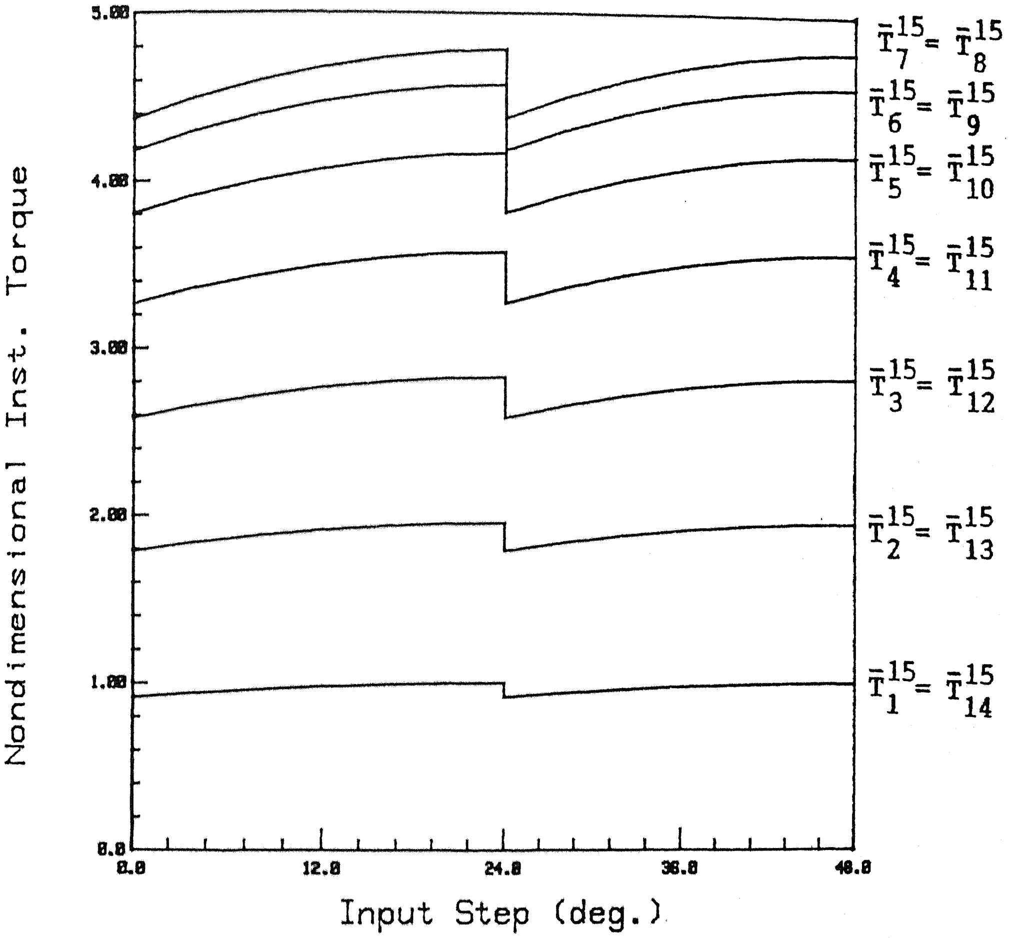


Figure 2.19 Non-dimensional instantaneous torque developed by the EHM - SM working in full-step regime ($N = 15$ and $1 \leq K \leq 14$)

$$\delta T_K^N = \frac{T_K^N\Big|_{max} - T_K^N\Big|_{min}}{2\, T_K^N\Big|_{avg.}} \qquad (2.61)$$

where $T_K^N\Big|_{avg.}$ is the average output torque of the motor which is given by :

$$T_K^N\Big|_{avg.} = \frac{T_K^N\Big|_{max} + T_K^N\Big|_{min}}{2} \qquad (2.62)$$

One important observation is that as $K \longrightarrow K^*$, though the amount of torque increases , the coefficient of torque pulsation also increases .

2.7.2 Theoretical Torque developed by the EHM - SM working in Half - Step regime.

The pressurization pattern sequence for the half-step regime is : $K \longrightarrow K+1 \longrightarrow K$ as shown in Fig. 2.20 . It is clearly seen that equation (2.57) still holds good for $\phi = 0$, $\phi = \alpha/2$ and $\phi = \alpha$. For other values of ϕ , i.e $0 < \phi < \alpha/2$ and $\alpha/2 < \phi < \phi$, the instantaneous torque developed by the motor can be calculated using equation (2.60) , keeping in mind that α should be replaced by $\alpha/2$ and proper values of C_k^N have to be used as the pressurization pattern changes from K to K + 1 and then again from K + 1 to K, depending on whether K is odd or even . Fig. 2.21 shows the graphical representation of non-dimensional instantaneous torque developed by the motor working in the half-step regime for N = 15

The following observations can be made from the figure:

1) Torque produced changes as switching on takes place from first step to second step.

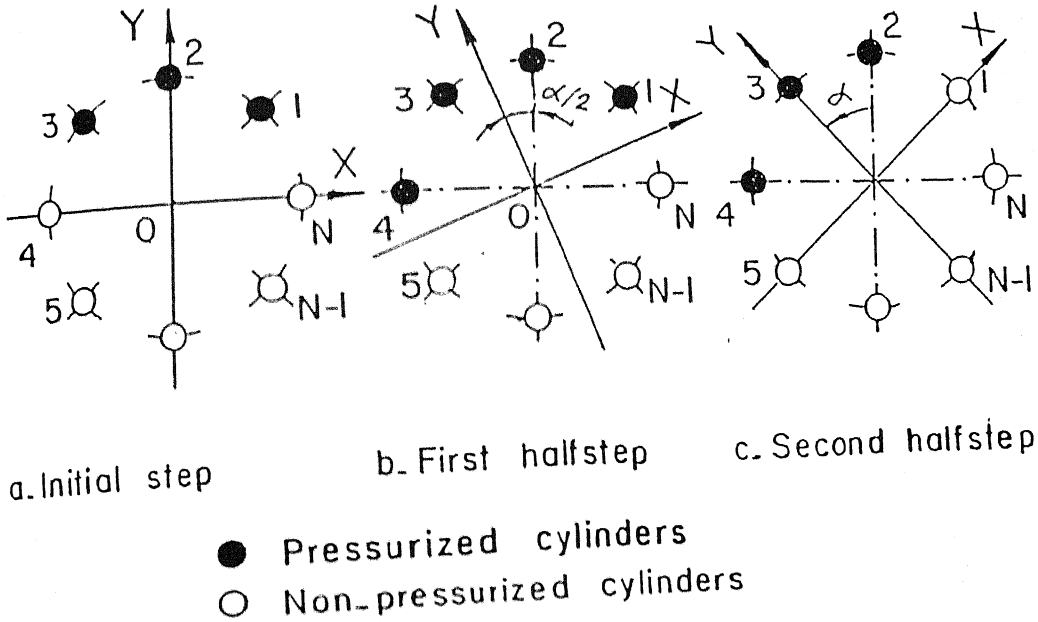


Figure 2.20 Pressurization pattern of cylinders working in half-step regime

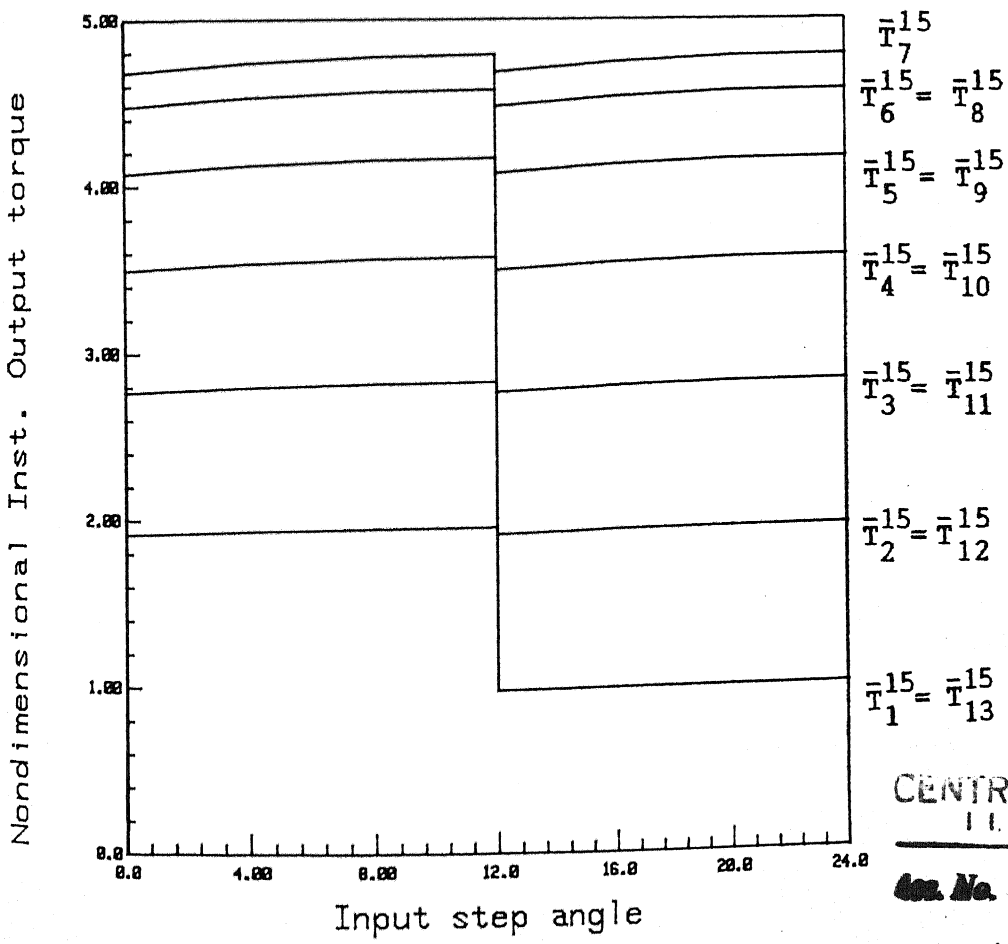


Figure 2.21 Non-dimensional instantaneous torque developed by the EHM - SM working in the half-step regime(N=15)

2) $\bar{T}_K^{NH} = \bar{T}_{N-(K+1)}^{NH}$

Superscript "H" denotes the torque developed for the half-step regime.

3) If $K < K^*$, then $\bar{T}_K^{NH} < \bar{T}_{K+1}^{NH}$

and if $K > K^*$, then $\bar{T}_{K+1}^{NH} < \bar{T}_K^{NH}$

2.7.3 Effect of supply pressure on output torque

The equation (2.57) can be written in the following form :

$$T_K^{NP} = \frac{r\ G \cos\gamma}{\tan\bar{\gamma}} \cdot \frac{\sum_{i=1}^{K} Y_i}{r}$$

$$= \frac{A\ P_S \cos\gamma}{\tan\bar{\gamma}} \cdot \sum_{i=1}^{K} Y_i \qquad (2.63)$$

The superscript "p" denotes that output torque is a function of supply pressure.

For a fixed set of geometrical parameters , e.g A , r , γ , $\bar{\gamma}$ and particular pressurization pattern , we have ,

$$\frac{A \cos\gamma}{\tan\bar{\gamma}} \cdot \sum_{i=1}^{K} Y_i = \text{constant} = C_K^{NP} \qquad (2.64)$$

$$\text{So} ,\ T_K^{NP} = C_K^{NP} \cdot P_S \qquad (2.65)$$

If only one cylinder is pressurized , the equation (2.65) becomes :

$$T_1^{NP} = C_1^{NP} \cdot P_S \qquad (2.66)$$

where $C_1^{NP} = \dfrac{A\ r \cos\gamma}{\tan\bar{\gamma}} = \text{constant}$

Let , $\bar{T}_K^{NP}$ be the non-dimensional holding torque defined as :

$$\bar{T}_K^{NP} = \frac{T_K^{NP}}{T_1^{NP}} = \frac{\sum_{i=1}^{k} Y_i}{r} \qquad (2.67)$$

$$= 1 + 2 \sum_{i=1}^{(K-1)/2} \text{Cos}\ (i\alpha) \qquad \text{when K-odd.}$$

$$= 2 \sum_{i=1}^{K/2} \text{Cos}\ (2i-1)\frac{\alpha}{2} \qquad \text{when K-even.}$$

Hence , taking equation (2.66) into account , we get ,

$$T_K^{NP} = C_1^{NP} . \bar{T}_K^{NP} . P_S \qquad (2.68)$$

For N = 15 , the above equation becomes :

$$T_K^{15P} = C_1^{15P} . \bar{T}_K^{15P} . P_S \qquad (2.69)$$

Figure 2.22 is a graphical representation of equation (2.69). The following observations can be made from this figure:

1) The output torque of the motor has a linear relation with the supplied pressure.

2) When no cylinder or all cylinders are pressurized , due to symmetry of pressurization pattern , no output torque is produced.

3) The output torque increases as $K \longrightarrow K^*$, either in the increasing or decreasing manner.

2.7.4 Effect of Pitch - cone angle on output torque

The effect of pitch-cone angle (θ) on the motor output torque can be demonstrated using equation (2.57). The equation can be written as :

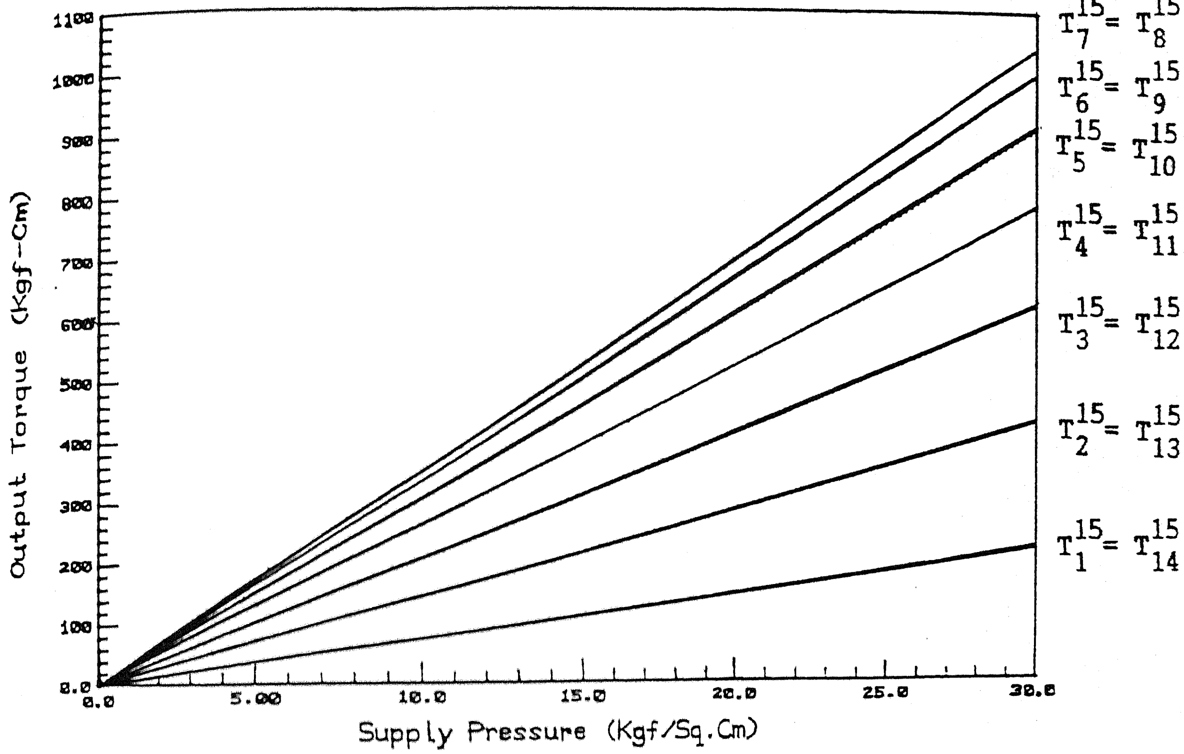


Figure 2.22 Effect of supply pressure on output torque

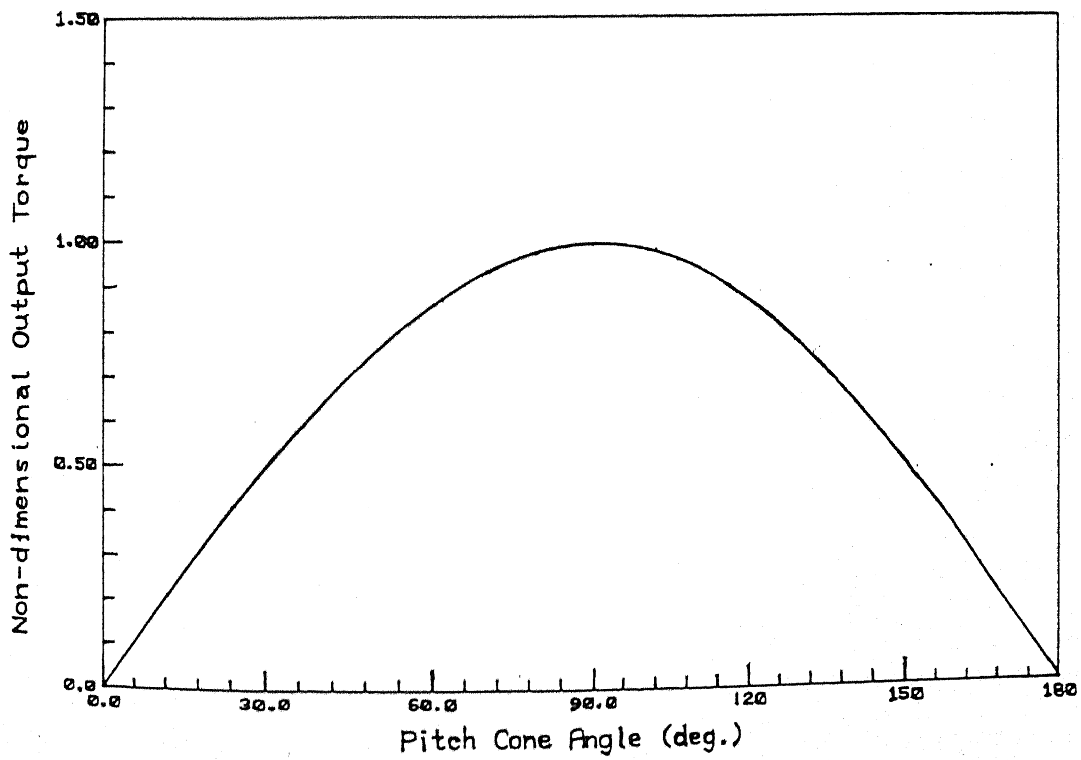


Figure 2.23 Effect of pitch - cone angle on output torque

$$T_K^{N\theta} = \frac{r\ G \cos\gamma}{\tan\bar{\gamma}} \cdot \frac{\sum_{i=1}^{K} Y_i}{r}$$

$$= \frac{A\ P_S \cos\gamma}{\tan\bar{\gamma}} \cdot \sum_{i=1}^{K} Y_i$$

$$= C \cdot \mathrm{Cos}\gamma$$

$$= C \cdot \mathrm{Cos}\ (90^0 - \theta)$$

$$= C \cdot \mathrm{Sin}\theta \qquad (2.70)$$

where $$C = \frac{A\ .P_S}{\tan\bar{\gamma}} \cdot \sum_{i=1}^{K} Y_i = \text{constant} \qquad (2.71)$$

Now , $$\frac{dT_K^{N\theta}}{d\theta} = C \cdot \mathrm{Cos}\theta \qquad (2.72)$$

Equating this derivative to zero , we get ,

$$T_K^{N\theta} = T_K^{N\theta}\Big|_{max} \text{ when } \theta = 90^0$$

So , $$T_K^{N\theta}\Big|_{max} = C \qquad (2.73)$$

Thus the equation (2.70) becomes :

$$T_K^{N\theta} = T_K^{N\theta}\Big|_{max} .\mathrm{Sin}\theta \qquad (2.74)$$

Non-dimensional torque of the motor is given by :

$$\bar{T}_K^{N\theta} = \frac{T_K^{N\theta}}{T_K^{N\theta}\Big|_{max}} = \mathrm{Sin}\theta \qquad (2.75)$$

Thus , the variation of non-dimensional torque with pitch-cone angle is in the form of a Sine-curve as shown in Fig. 2.23. The maximum torque occurs at $\theta = 90^0$ and minimum torque occurs at $\theta = 0^0$ or 180^0. Since $\gamma = 90^0 - \theta$, the tilt angle (γ) should be as small as possible.

CHAPTER - 3

DESIGN PROCEDURE OF THE PROPOSED EHM - SM

3.1 Introduction

The design procedure of the proposed EHM-SM is relatively simple. As far as hydraulic part of the motor is concerned , it differs from ordinary hydraulic motors in that instead of the valve being stationary and cylinder block rotating , in this case , the valve is rotary and cylinder block is stationary. The fixed and nutating gears are standard bevel gears whose design procedures are known. The rotating bevel gear has its pitch slightly different from that of the nutating gear — the former has one tooth more. The important requirement is that of tooth profile , especially the pitch of the gear that has to be made with a high degree of accuracy. Otherwise , it will affect the built-in-step accuracy of the motor. Some important steps of design procedure have been taken into account in this chapter. Besides , design procedure such as steps to design the valve and torque required to rotate the valve is also presented in this chapter.

3.2 Required design information

To design the proposed EHM-SM , the following informations are required :

1. Desired motor characteristics :

   a) Step size (β)

   b) Stepping speed (f)

   c) Holding torque

   d) Allowable coefficient of torque fluctuation (δT^N)

2. Operating conditions :

   a) Maximum allowable working pressure (P_S)

b) Occupied space and method of coupling with other equipment.

c) Ambient conditions like temperature , humidity etc. in which motor works.

3.3 Important steps of design procedure

The following steps have been observed :

I. Determination of the total numbers of cylinders (N).

Data required : Amount of holding torque to be developed , keeping in mind that the optimum number of cylinders are pressurized.

Procedure :Calculate N using equation (2.57). Total number of cylinders is such that the center distance between two successive should not be less than 1.3d , where d is the diameter of the cylinder. Otherwise , leakage may result. N is recommended to be rounded off to the nearest odd integer.

II. Determination of optimum number of pressurized cylinders (K^*)

Data required : Total number of cylinders (N) selected in step I.

Procedure :Calculate K^* using relation $K^* = \frac{N \pm 1}{2}$ when N is odd and $K^* = \frac{N}{2}$ when N is even. Generally odd number of cylinders (N) is preferred.

III. Determination of the number of teeth for fixed , nutating and rotating bevel gears (n_F , n_N , n_R).

Data required : Step size (β) , total number of cylinders (N) selected in step I.

Procedure :

1) Try to assume least teeth difference between rotating

and rotating bevel gears i.e $(n_R - n_N) = 1$.

2) Calculate n_R using equation (2.8).

3) So , n_N can be calculated using relation $n_N = n_R - 1$

IV. Selection of tilt angle (γ) of the nutating gear .

Data required : Non - dimensional torque developed by the motor.

Procedure : Select tilt angle (γ) using equation (2.75) and keeping in mind that $\gamma = 90^0 - \theta$

V. Selection of piston diameter (d)

Data required :

1) Maximum allowable pressure (P_S)

2) Required holding torque

3) Tilt angle (γ) selected in step IV

4) Standard pressure angle , $\bar{\gamma} = 20^0$

5) Optimum number of pressurized cylinders (K^*).

6) Total number of cylinders (N) selected in step I .

Procedure :

1) Calculate C_K^N using equation (2.58) or (2.59) , depending on whether K is odd or even .

2) Using the value of C_K^N , pitch circle radius of cylinders (r) can be computed from equations (2.54) & (2.57) . Tilt angle (γ) , selected in step IV in this equation .

3) Piston diameter (d) can be computed with the help of relation : $d = \dfrac{2 \Pi r}{1.3 N}$

VI Determination of effective axial travel (S_K^N) of pistons .

Data required :

1) Radius of pitch circle of cylinders (r)

2) Tilt angle (γ) selected in step IV.

4) Total number of cylinders selected in step I .

Procedure : Calculate S_K^N using equation (2.16).

VII Determination of average theoretical flow rate (Q_K^N) required by the motor.

Data required :

1) Stepping speed (f)

2) Computed S_K^N from step VI.

Procedure : Calculate Q_K^N or $\bar{Q}_K^N$ using equation (2.17) or (2.18)

VIII Determination of valve-sleeve port dimensions.

Data required :

1) Optimum number of pressurized cylinders (K^*) in one step calculated in step II.

2) Theoretical average flow rate (Q_K^N) calculated in step VIII .

3) Allowable fluid velocity (V_1) in the valve-sleeve port.

Procedure :

1) Calculate the area of the valve-sleeve (A_p) as :

$$A_p = \frac{Q_K^N}{K^* . V_1}$$

2) The valve-sleeve port is rectangular . If the port width is (b) and the port length is (a) , then :

$$A_p = a . b$$

3) Assigning any suitable value to port width (b) [depending on the diameter of end mill] , port length (a) can be calculated.

IX Determination of valve diameter (d_v)

Data required :

1) Valve-sleeve port width (b).

2) Distance (c) between two successive rectangular ports of the valve-sleeve.

3) Total number of cylinders (N) selected in step I.

Procedure :

1) Assume $c = 1.5\ b$

2) Calculate the valve diameter (d_v) using the relation :

$$d_v = \frac{N\ (\ b + c\ )}{\Pi}$$

The valve diameter d_v should be rounded of to the nearest integer .

X Design of valve land

Data required :

1) Total number of cylinders (N).

2) Optimum number of pressurized cylinders in one step (K^*)

Procedure :

1) Calculate valve land width (h) , assuming over-lapped valve i.e $\delta h = h - b > 0$. Actually valve land width (b) should be such that $b \leq h \leq c$.

2) Calculate the valve land location using equation (2.10) or (2.12), depending on whether the valve is of type I or of type II .

3.4 Theoretical torque required to rotate the valve.

The following assumptions are made :

1) Weight of the valve is negligible .

2) Pressure gradient in radial clearance between the valve and the valve sleeve is negligible.

The viscous torque (T_v) resisting the rotation of the valve is given by :

$$T_v = F_v \cdot \frac{d_v}{2} \qquad (3.1)$$

where, F_v = Viscous force.

and d_v = Diameter of the valve .

According to Newton's law of viscosity , viscous force (F_v) is given by :

$$F_v = \tau \cdot W = \mu \cdot \frac{du}{dy} \cdot W \qquad (3.2)$$

where du = Relative peripheral velocity between valve and sleeve.

dy = Radial clearance between valve and sleeve .

μ = Dynamic viscosity of fluid.

τ = Shear stress of fluid .

W = Surface area of valve = $\Pi\, d_v\, L$.

L = Length of the valve .

$$\text{Now , } du = \frac{\omega\, d_v}{2} \qquad (3.3)$$

$$\text{and } \quad dy = \delta_c \qquad (3.4)$$

= Radial clearance between valve and sleeve .

Thus , using equations (3.1) , (3.2) , (3.3) and (3.4) , the equation (3.1) becomes :

$$T_v = \mu \cdot \frac{\omega\, d_v}{2} \cdot \frac{1}{\delta_c} \cdot \Pi\, d_v\, L$$

$$= \frac{\mu\ \Pi^2\ d_v^2\ L}{N\ \delta_c} \cdot f \qquad (3.5)$$

Equation (3.5) shows that to minimize the torque required to rotate the valve , its length and diameter should be as small as possible. This is in sharp contrast with Loc's valve[19]. Our proposed motor uses the valve having diameter and length both equal to about half of the corresponding dimensions of Dr. Loc's valve , although torque developed is of greater amount in the present case.

Substituting the actual parameters of the prototype EHM-SM ,i.e N = 15 , L = 36.5 x 10^{-3} m. , d_v= 12 x 10^{-3} m

and $\delta_c = 5 \times 10^{-6}$ into equation (3.5) , we get ,

$$T_v = 0.691 \, \mu \, f \qquad (3.6)$$

3.5 Salient features of prototype design of the proposed EHM - SM

Our objective is to design , fabricate and test a prototype of the proposed EHM-SM .

The design informations have been given taking following considerations into account :

1) The available input motor and hydraulic power package.

2) Manufacturing and assembling facilities available in central workshop and manufacturing science laboratory of I.I.T. Kanpur.

Our principal area of concentration has been on two items - output step angle equal to 0.489^0 and holding torque equal to 870 kgf-cm. at $P_s \leq 25$ kgf/cm^2.

The primary steps of design procedure have been presented step-wise in section 3.3 of this chapter. However, to reduce manufacturing work and at the same time to achieve smooth and perfect functioning of the proposed motor, some minor compromisation has been made. As for example, making rectangular slots of 3.5 mm. × 1 mm. and having depth of 6 mm in copper valve sleeve with the help of end mill was extremely difficult. So, to get rid of the problem, drilling was made at three positions and then with the help of needle file the slot has been cleared off. Besides, radial and axial dimensions of valve, diameter of the piston has been increased than actually needed. Contact between piston and mutating bevel gear has been reduced to point contact with

certain amount of radius of curvature so that wear action is minimized and at the same time operation is smooth.

The prototype design has been made in such a way that external leakage from the cylinder block and that from the valve can be collected separately . The length and diameter of tubes (connectors) for the tube casing have been selected in such a way that there is no leakage of fluid. Otherwise , supply pressure will fall and affect the performance of the motor considerably. The coupling between ESM and the valve is rigid in radial direction but compliant in axial direction to take into account any mis-alignment between the two units. Table 3.1 shows the major geometrical parameters of the proposed motor.

In this chapter , a number of coloured photographs has been presented to get a proper understanding of the units developed. The motor in test bed is shown in Fig. 3.1 . For easy detection of parts , a comprehensive view of all the parts of the proposed motor is shown in Fig. 3.2 . The combination of three gears - fixed , nutating and rotating is shown in Fig. 3.3. The cylinder block , tube casing and pistons are shown in Fig. 3.4. The valve-sleeve combination is shown in Fig. 3.5 .

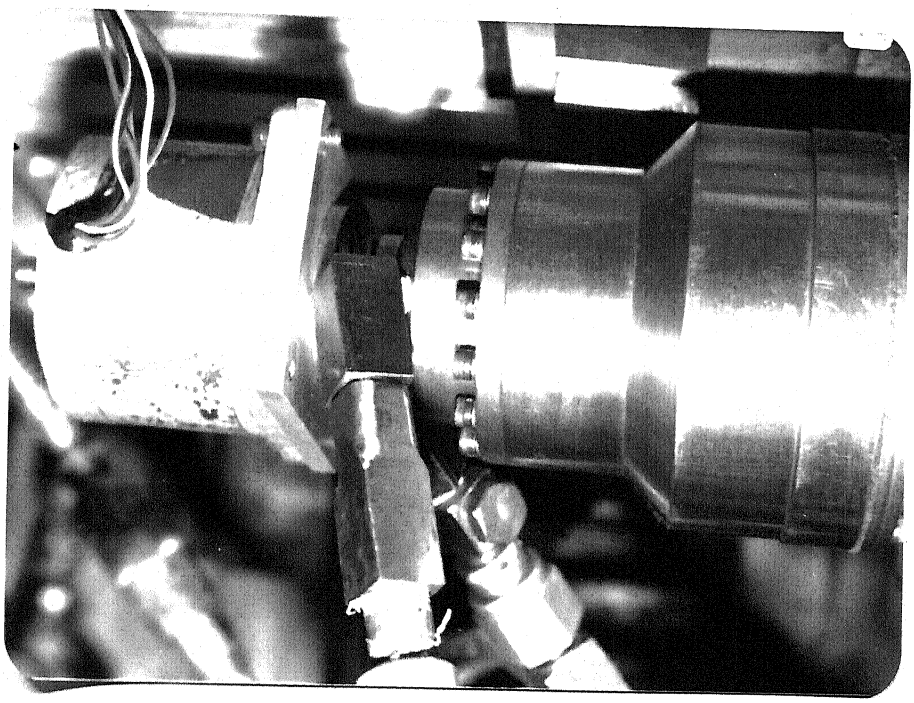


Figure 3.1 Protoyype EHM - SM on the test bed

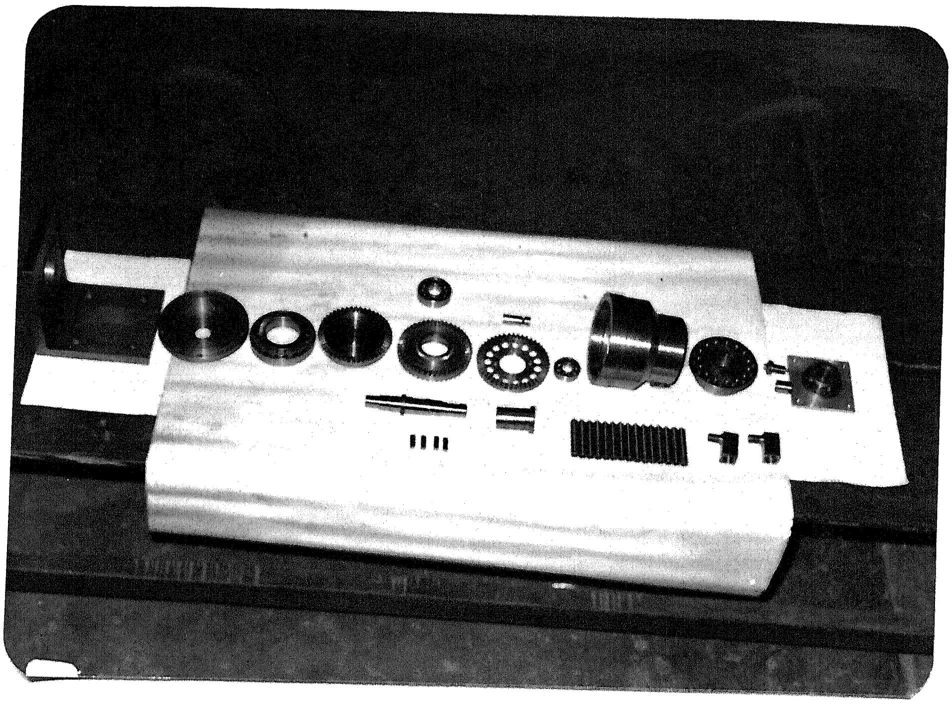


Figure 3.2 Comprehensive view of the parts of EHM - SM

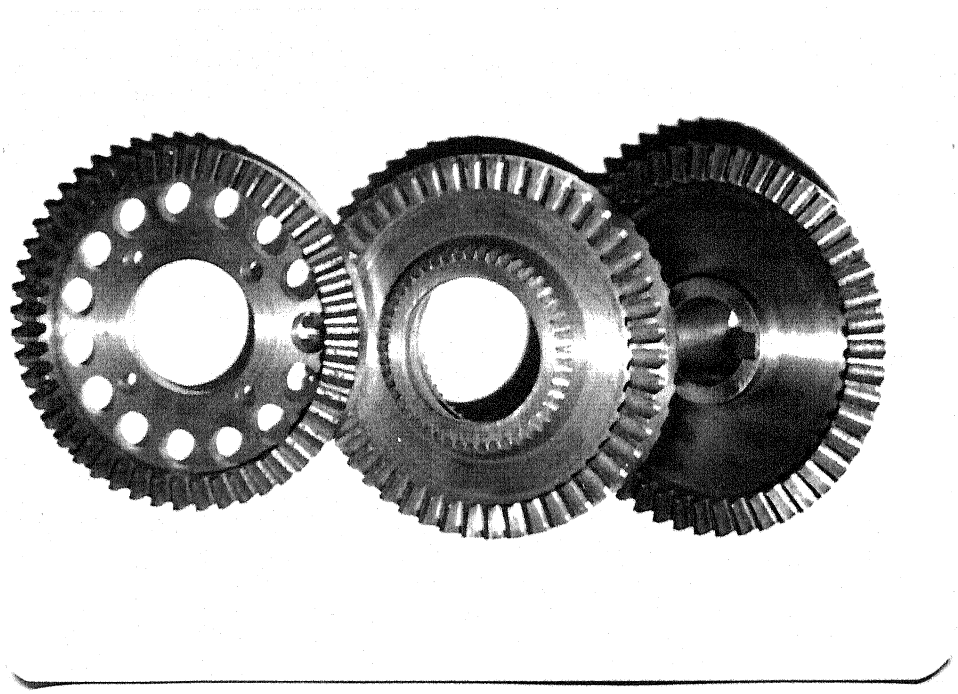


Figure 3.3 Combination of three bevel gears

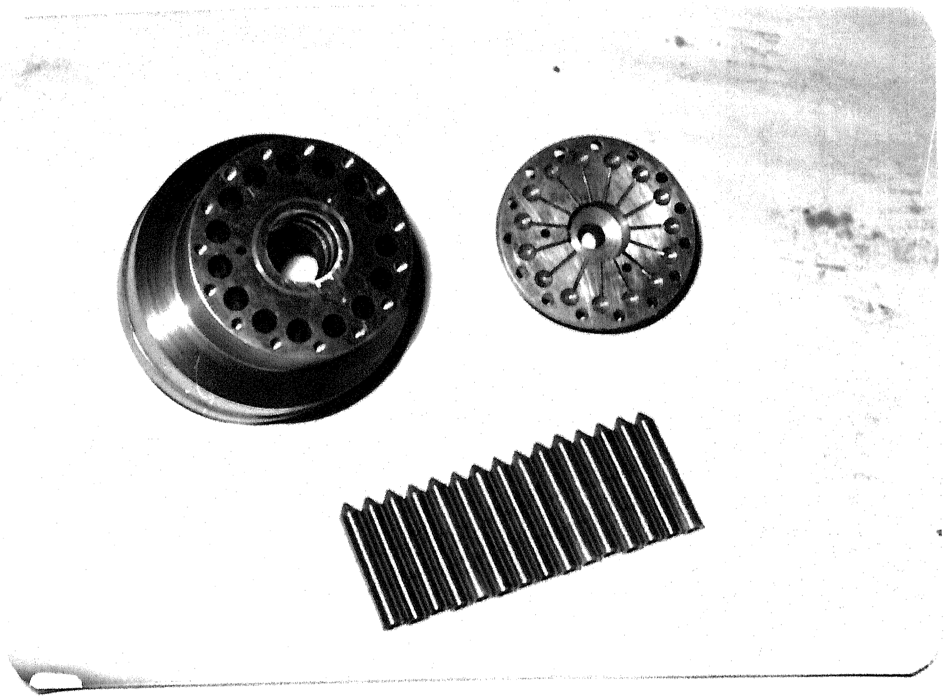


Figure 3.4 Cylinder block , tube casing and pistons

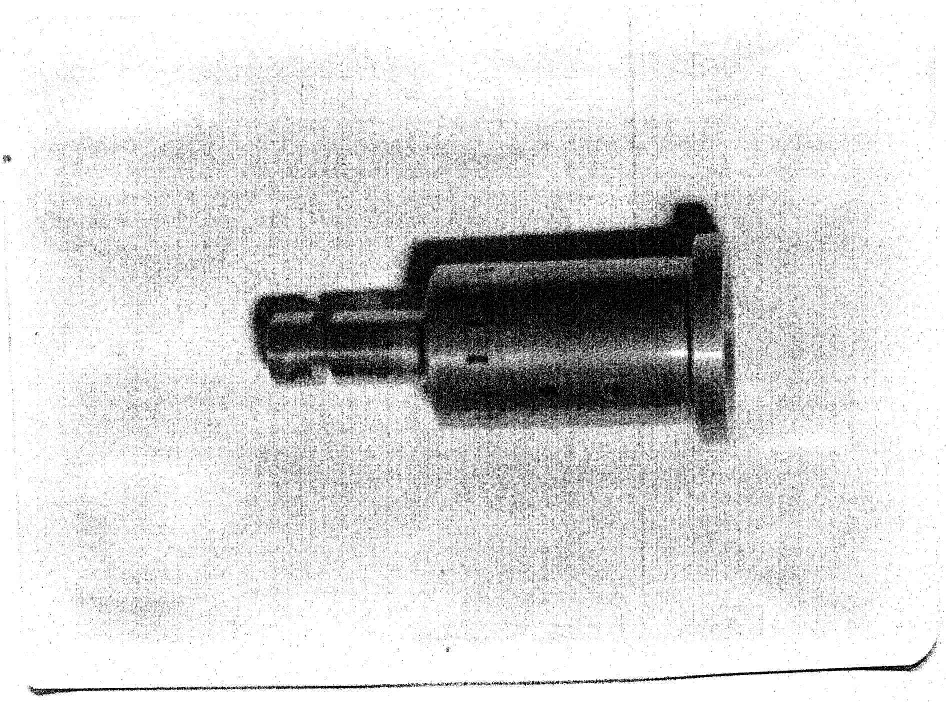


Figure 3.5 Valve - sleeve combination

Number	Parameters	Values
1.	Input step angle (α)	24^0
2.	Output step angle (β)	0.489^0
3.	Tilt angle of nutating gear (γ)	7^0
4.	Pitch-cone angle (θ)	83^0
5.	Gear pressure angle ($\bar{\gamma}$)	20^0
6.	Total number of cylinders (N)	15
7.	Optimum no. of pressurized cylinders(K^*)	7
8.	No. of teeth of fixed gear (n_F)	48
9.	No. of teeth of nutating gear (n_N)	48
10.	No. of teeth of rotating gear (n_R)	49
11.	Diameter of piston (d) (mm)	10
12.	Length of piston (mm)	60
13.	Valve diameter (mm)	12
14.	Length of valve (mm)	36.5
15.	Diameter of output shaft (mm)	18
16.	Piston-cylinder clearance (mm)	5×10^{-3}
17.	Valve-sleeve clearance (mm)	5×10^{-3}
18.	Module of gears (mm)	2
19.	Maximum diameter of motor (mm)	121
20.	Length of the motor (mm)	225

Table 3.1 - General parameters of the prototype of EHM -SM

CHAPTER 4

TESTING AND PERFORMANCE OF THE PROTOTYPE EHM - SM

4.1 Experimental set-up

The experimental set-up is shown schematically in Fig. 4.1. The set-up consists of (1) electrical stepper motor, (2) prototype EHM-SM, (3) control box , (4) pony brake, (5) hydraulic power supply unit, (6) spring scale, (7) pressure gauge, (8) throttle valve and (9) distributor (start/stop valve). A brief information of some of the important components are given below :

4.1.1 Electrical stepper motor

ESM used in the experimental set-up is "Sri - Syn" d.c. stepping motor which converts the digital electronic signal into precise angular incremental motion. "Sri - Syn" d.c. stepping motors are center tapped, bifiler type motors with six leads i.e, four phases and two commons. The step angle is 1.8 deg. per full step and 0.9 deg. per half step. The different models of "Sri - Syn" d.c. stepping motors range from Type STM 601, 2 kgf-cm torque to Type STM 1103, 60 kgf-cm torque. The particular ESM used in this experimental set-up has the following specifications:

Sri - Syn d.c. stepping motor

Type - STM - 901

Sr. No. - 67 - 2

Input 12 volts d.c.

0.67 amp/phase

Torque - 7 kgf-cm

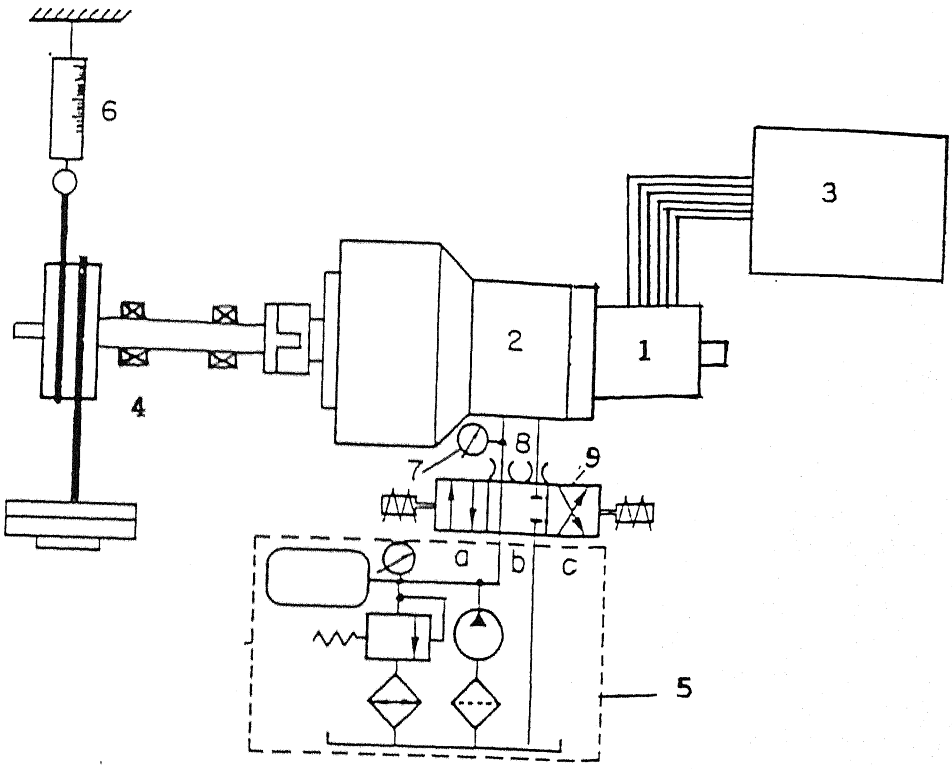


Figure 4.1 Schematic diagram of test - bed

4.1.2 Control box

The control box houses (I) electronic step motor logic controller and (II) Power drive.

Electronic step motor logic controller :

The four-phase switching logic is generated here in appropriate sequence and is supplied to the power drive. The controller has the facility for selection of different modes of operations. These modes are panel or remote control , index or continuous operation , full step (1.8^0) or half step (0.9^0) motion. There is also a facility to set the speed of the d.c stepping motor in number of steps/second , maximum up to 4 digits.

Power drive :

"Sri - syn" power drive is basically a power supply unit. The power required for the stepping motion is supplied to the motor phases in a sequence generated by the controller. The power switches for the motor phases are also housed in this unit. The switching logic supply by the controller is amplified and then the switching power transistors are accordingly driven. The specified torque of the d.c stepping motor is the torque in the holding condition and is proportional to the current in the motor phases. There are three types of power drives :

(a) Type : R/L :-

The R/L type power drive is simple and economical but has limitations on the speed of the d.c stepping motor. The R/L drive can maintain the torque up to about 150 to 200 steps per second.

(b) & (c) Unipolar chopper & Bipolar chopper :

The unipolar chopper & bipolar chopper drives are comparatively expensive but have advantages of d.c stepping motor operations with workable torque up to 3000 steps/sec. (900 r.p.m). These

power drives use Thyristorised circuitry. The power required at different speeds is accordingly pushed in the motor coils by controlling the Thyristor firing angle. Thus, these power drives improve the dynamic performance ,i.e. the torque/speed characteristics of the d.c. stepping motor.

The bipolar chopper drives are available for the d.c stepping motors , type STM 601 , STM 602 , STM 603 , STM 901 with torque up to 2 , 4 , 6 , 7 Kgf.cm respectively.

The unipolar chopper and R/L power drives are available for complete range of "Sri - Syn" d.c. stepping motor, type STM 601 (2 kgf-cm torque) up to type STM 1103 (60 kgf-cm torque).

4.1.3 Pony Brake

The pony brake consisting of (a) a spring scale (manufacturer - Salter, made in India, Load - 25 kg.), (b) a canvass belt, (c) a drum and (d) a collection of pan weights as shown in Fig. 4.1. In the present set up, the pony brake is used to create frictional load torque acting on the output shaft of prototype motor. Since the diameter of the drum is 17 cm., the frictional load torque (T_L) acting on the test motor shaft is given by :

$$T_L = 8.5\ (M - \bar{S}_r)\ \text{kgf-cm.}$$

where M = mass of the pan, kg.

and $\bar{S}_r$ = reading of the spring scale, kg.

4.1.4 Hydraulic power supply unit

Specification :

Manufacturer : Vicker Sperry of India Limited

Model : P.U.10132A, Pressure range : 0 - 40 kgf/sq.cm.

Flow rate : maximum 20 decimeter^3/minute

The hydraulic unit supplies fluid (SAE 30,viscosity μ = 4×10^{-2} N-s/m^2) to the test motor through a distributor (9) as

shown in Fig. 4.1. The temperature of the working fluid can be maintained at a desired level by use of the built-in water to oil heat exchanger.

4.1.5 Distributor

The function of the distributor is to supply pressurized fluid to the forward chamber of the test motor and to make a way for depressurized fluid from the backward chamber back to the sump. The distributor is basically a 3-position 4-way valve. The three possible positions are - (a) advance, (b) neutral and (c) retreat. In advanced position, the pressurized fluid is supplied to the forward chamber, while in retreat position the backward chamber is pressurized. The distributor, thus, can also change the direction of rotation of output shaft of prototype motor. For the neutral position of the distributor, neither the forward chamber nor the backward chamber is pressurized.

4.2 Pre-experimental run of prototype motor

After the proposed motor is fabricated, assembled and installed in the set-up shown as in Figure 4.1, it has been subjected to a rigorous trial governed by various rules of the acceptance test of hydraulic machines. Under no load condition, the proposed motor was run for twelvehours in either direction of rotation and in different stepping speeds in full step as well as half step regimes. The proposed motor also works successfully with wide range of external inertia loads. The pre-experimental run also includes twenty four hours of demonstrative running in front of many enthusiastic visitors and students.

4.3 Experimental results and discussions

The following two experiments have been carried out :

1.) Measurement of the holding torque of the motor.

2.) Measurement of accuracy and consistency of steps.

4.3.1 Measurement of holding torque

The objective of this experiment is to plot the holding torque of the prototype motor as a function of supply pressure. The experimental set-up is shown in Fig. 4.2.

The following steps are followed :

1) Stop the input ESM so that there is no rotary motion of the valve.

2) Switch on the distributor lever either to advance (a) or retreat (c) position.

3) Set the desired supply pressure by adjusting the throttle valves.

4) Load the output shaft of the test motor by putting known weights onto a pan attached to the belt hanging from the drum which is rigidly mounted on the shaft. Loading is continued till the motor shaft rotates due to the external weights.

5) Set a new supply pressure and repeat the same process.

The maximum frictional load torque , the test motor could take without losing its step is called the holding torque. The registered values of holding torque and supply pressure are shown in Table 4.1.

The holding torque, as a function of supply pressure, is shown in Fig. 4.3. The figure shows that the discrepancy between the theoretical and experimental values vary between -5% and +5%.

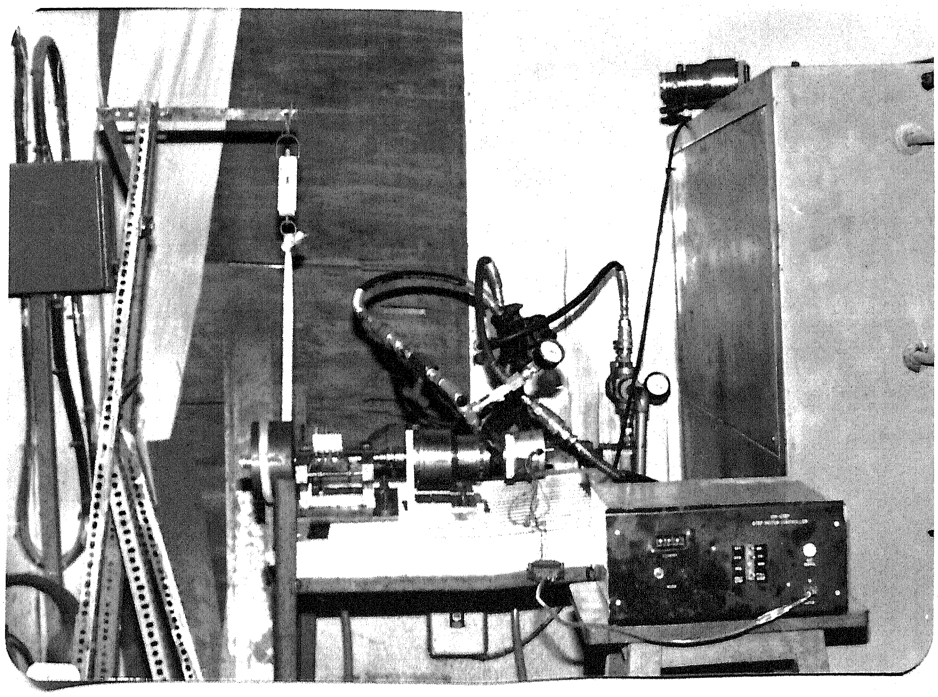


Figure 4.2 Experimental set-up for measurement of holding torque

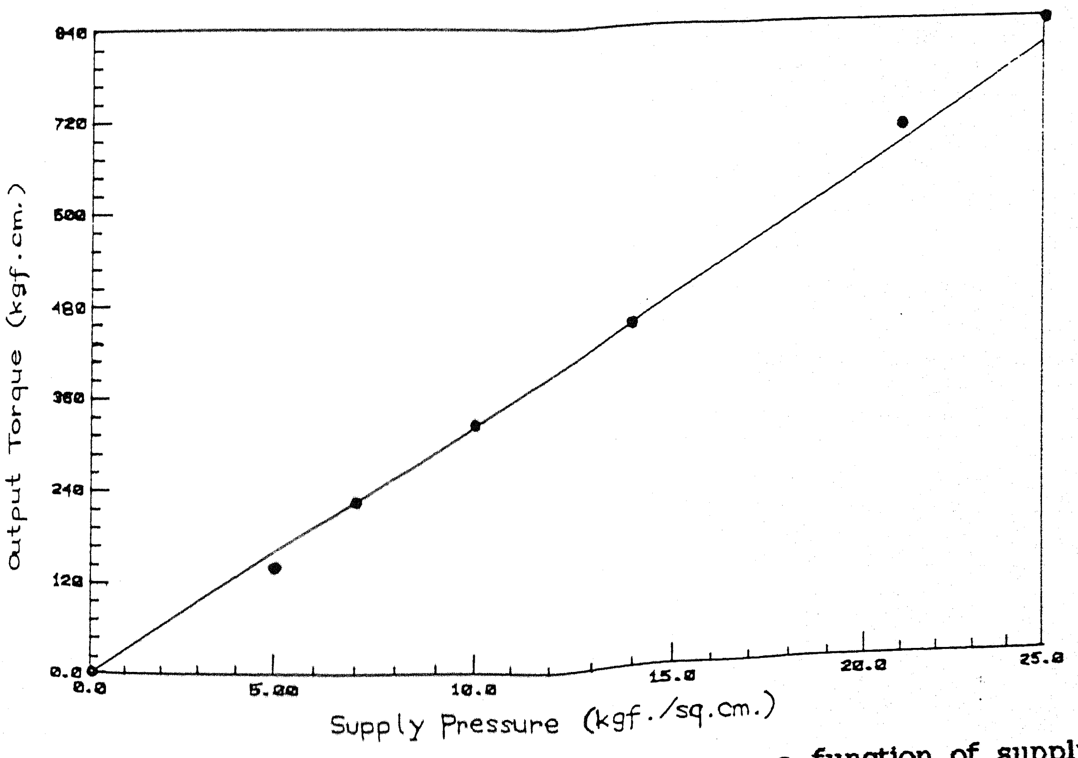


Figure 4.3 Measurement of holding torque as a function of supply pressure

Supply pressure (kgf/cm^2)	Holding torque (kgf-cm)
0	0
5	140
7	225
10	325
14	450
21	700
25	840

Table 4.1

4.3.2 Measurement of accuracy and consistency of step size

The experimental set-up is shown in Fig. 4.4. Calibration of the input electrical stepping motor is done before the steps are measured.

The calibration of ESM is shown in Fig. 4.5. Now, for ESM, we have: steps per second X time = number of steps. Since the time and number of steps in terms of number of revolution can be measured with the help of a stop-watch and a tachometer, the actual steps per second can be calculated. If the set steps per second in the controller is represented as Y and the actual steps per second is represented as X, then the following relation is obtained: $X = 0.5\ Y$

Set-up description:

A circular aluminium plate is rigidly mounted on the output shaft of the proposed motor. A white paper is pasted on the plate. A working pointer is fixed on the test bed with the help of a guide

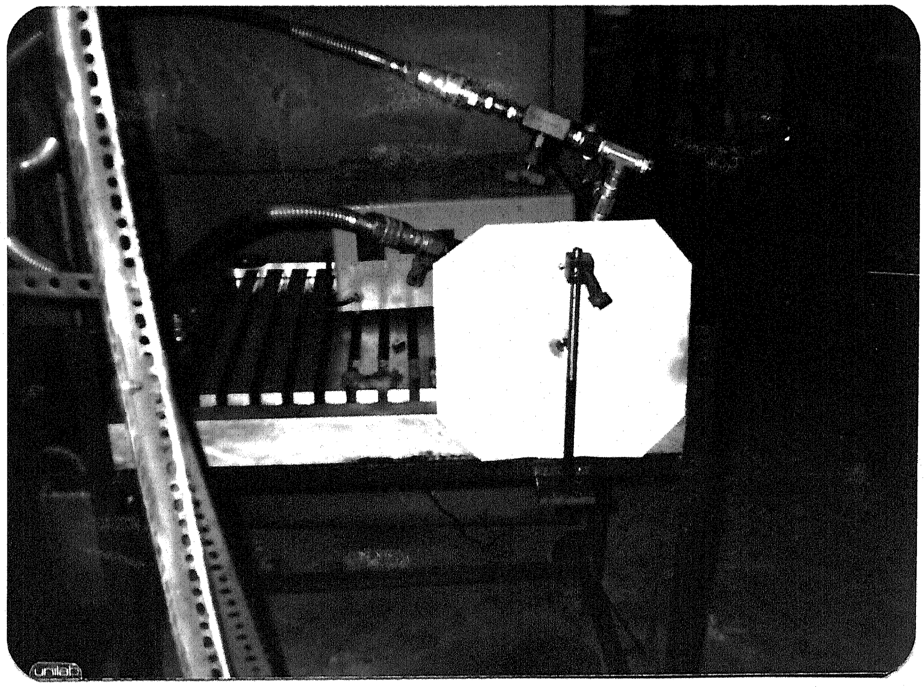


Figure 4.4 Experimental set-up for the measurement of steps

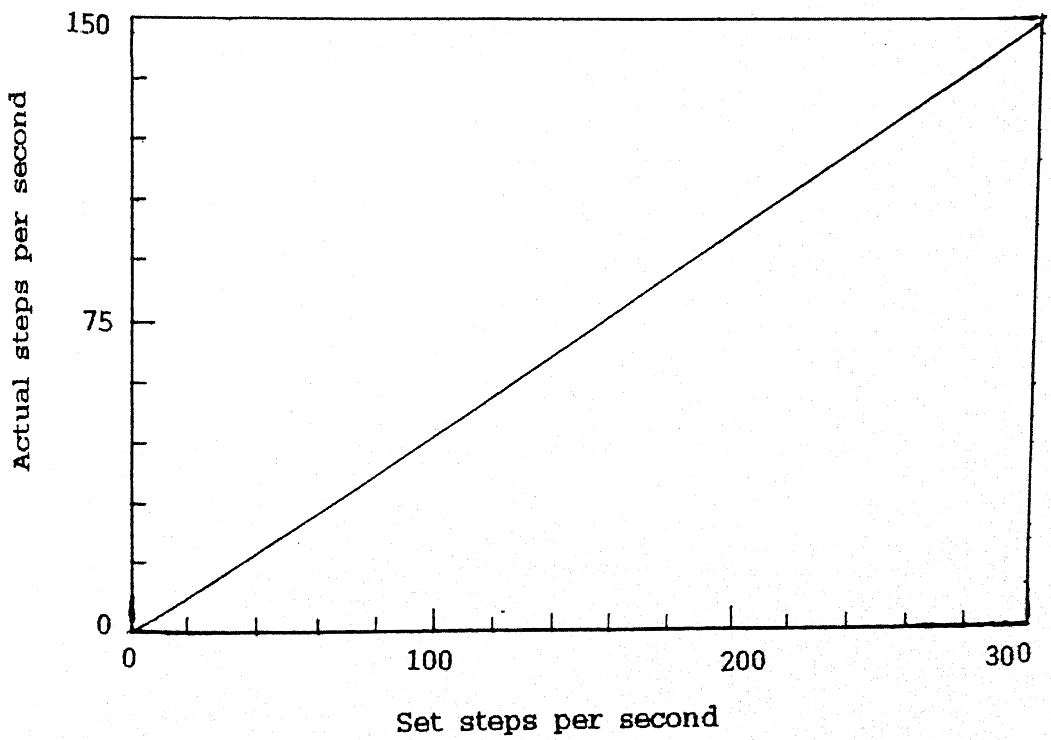


Figure 4.5 Calibration of ESM

block and the needle of the pointer is directed vertically to the circular aluminium plate. In each step of rotation of the output shaft and the aluminium plate alongwith it, the needle pointer is pushed towards the plate and the step is marked.The plate should be sufficiently thick so that it cannot bend in the vertical plane about its axes when small point load is applied during the marking of steps.

The following steps are followed:

(i) Checking is done with the help of marking block whether the plate is perpendicular to the output shaft axis while rotating.

(ii) Full step of the proposed motor = $\frac{24}{49} = 0.489^0$. So, one complete revolution of the plate should consist $\frac{360 \times 49}{24} = 735$ steps. The pointer is fixed on a circle having diameter equal to 234 mm so that pitch of the steps should be equal to $\frac{\Pi \times 234}{735} =$ 1 mm . The pointer may be fixed at any point on the circle taking it as a reference point.

(iii) Set a suitable supply pressure.

(iv) One full step of ESM = 1.8^0. One input step angle $\alpha = 24^0$ gives one output step. If one step per second is set on the controller, then time elapsed between two successive steps is equal to $\frac{24}{1.8} = 13.3$ seconds which can be conveniently measured with the help of a stop watch.

(v) Therefore, for every 13.3 seconds one output step is generated and the spring loaded pointer is pushed onto the plate to mark the step.

(vi) The procedure is continued till one revolution of the plate.

The measurement of steps is shown in Fig. 4.6. The pitch of the step is consistent and is equal to 1 mm except that for every 15

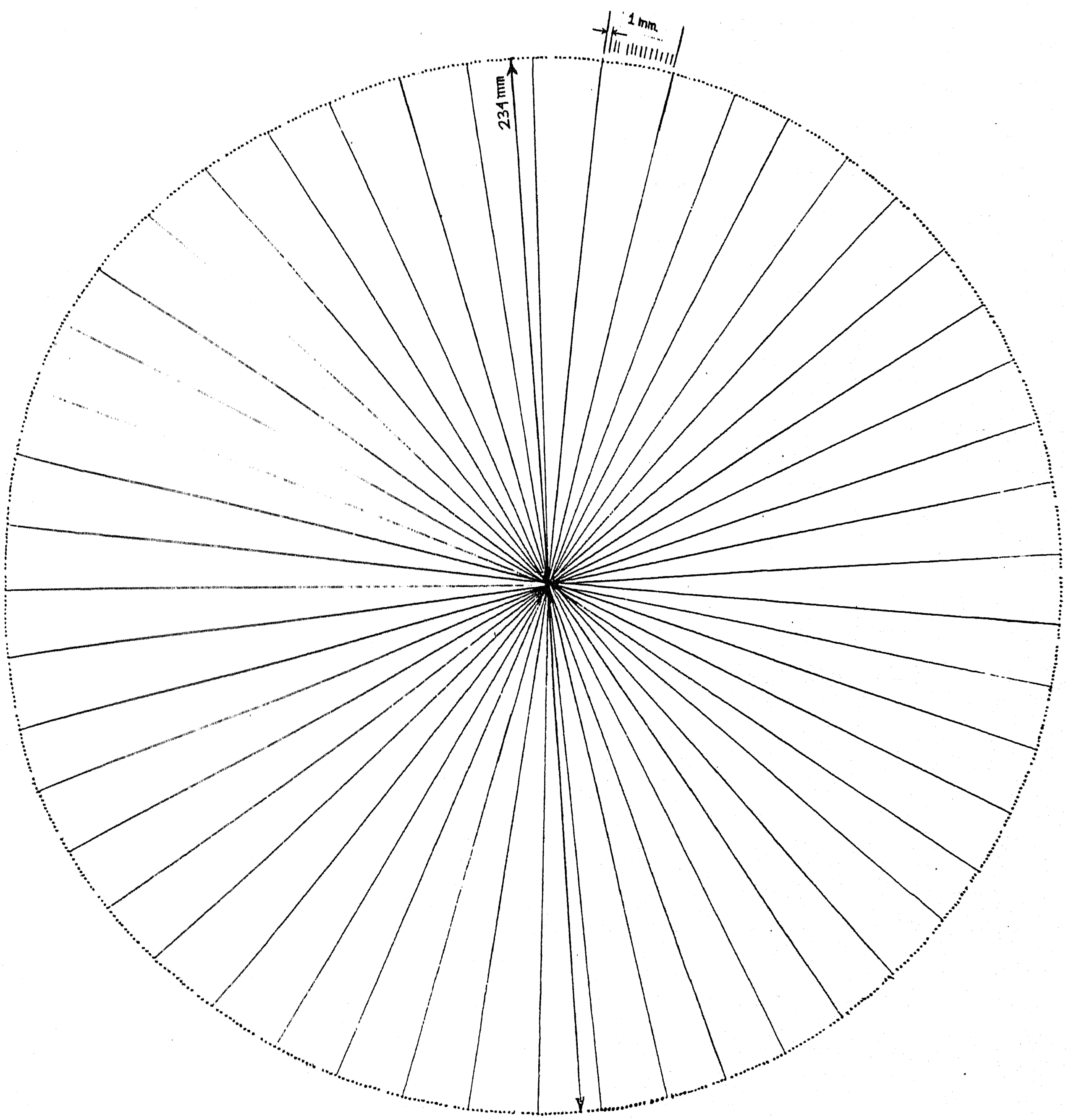


Figure 4.6 Measurement of steps

steps one step gets lost. This can be seen from the step marks that there is a gap of about 2 mm for every 15 steps. Actually 15 output steps are generated for one complete cycle. i.e for one full participation of 15 cylinders. One probable reason for step loss is that one particular piston is not being pressurized. So, the resultant force which is responsible for producing steps jumps one step ahead. The other probable reason may be due to the incorrect pitches of the nutating-rotating gear combination. The problem can be avoided by properly lapping holes inside the cylinder block and accurate manufacturing of bevel gears.

4.4 Conclusions

The indigenous design of new model of electro-hydro-mechanical stepping motor has been proposed and a prototype has been fabricated. This prototype EHM-SM overcomes the drawbacks of the existing stepping motors. This motor has the unique capacity of producing 735 steps per rotation coupled with a considerable amount of torque of 840 kgf-cm at 25 kgf/cm^2 pressure.

The important conclusions of the work are :

A - The new model of EHM-SM possesses the following features ;

1. Simple design.
2. High torque.
3. High resolution.
4. High power-to-weight ratio.
5. Act as a speed reducer and torque amplifier.

B - In order to produce uniform output torque , this prototype EHM-SM has the following design considerations :

1. An odd number of cylinders (e.g N = 15)
2. Pressurization of optimum no. of cylinders (e.g K = 7)
3. Type - II valve. [$\Psi_1 = (2k + 1)\frac{\alpha}{2}$]

4.5 Suggestions for future work

Though sufficient efforts have been given to design and fabricate the motor, there is enough scope to improve its working condition. The special area of attention should be the cooling and lubrication system of the motor. Amount of leakage of the motor can be reduced with proper surface finishing operations. As far as built-in step accuracy of the motor is concerned, gears are the critical parts that has to be manufactured properly. Only two principal experiments are carried out. The following experiments can be performed by future researchers to reinforce the claim of the theoretical results.

1.) The motor velocity characteristics.

2.) The motor dynamic characteristics.

3.) Actual coefficient of torque fluctuation of the motor.

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